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WINNING THE ENGINEERING TROPHIES, 1924-1925.

By COMMANDER R. R. SMITH, U. S. NAVY, MEMBER.*

PART I.

THE ENGINEERING COMPETITION.

The engineering performance year 1924-1925 was marked by intense competition between the leading vessels in the various classes. It must be assumed that the allowances for the various ships are equitably established and that the differences in scores represent the differences in individual fuel consumptions when compared to the individual standards assigned. Inequalities in allowances did undoubtedly exist, but these inequalities may be considered to have been equally distributed

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and therefore neutralized. A score attained by a vessel must, therefore, be considered as a function of the engineering performance of the vessel and the vessel attaining the highest score in its class must be considered as having attained the highest excellence in engineering performances in its class.

Naval officers are familiar with the methods and details of the engineering competition, but many members of this Society and readers of this Journal are not acquainted with the basic principles upon which the competition is conducted. Prior to discussing the methods used by the trophy winners, a brief explanation will be given of the engineering competition as conducted in the Navy in order that the subject may be more readily understood. In order to sum up the daily, monthly and yearly engineering performances of a vessel and to compare vessels in the same class, it is necessary to provide a method of scoring. A vessel's service demands essential changes in the disposition of the plant for the different conditions of operation. Each condition calls for a different rate of expenditure of fuel. There are two general conditions under which a vessel operates—namely, underway and not underway. These conditions for oil burning vessels are subdivided as follows:—

(*a*) Performances underway.

(*b*) Performances not underway.

(1) At anchor.

(2) Standing by (awaiting engine bells from the bridge).

(3) Operating without boilers (receiving steam).

A vessel must operate under one of the above conditions, and standard hourly fuel allowances for each condition are assigned to each vessel before the commencement of the competition year. These allowances or standards are based on actual past performances and are established, not on the average performance, but upon the average of the *best* recent performances.

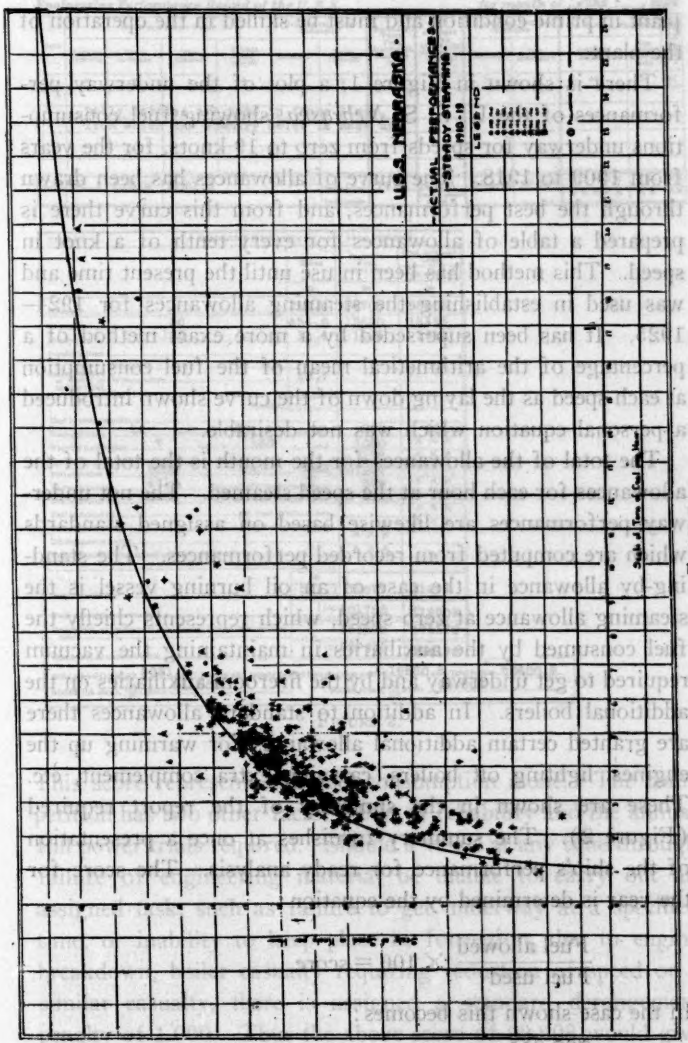

U.S.S. NEBRASKA.
ACTUAL PERFORMANCE
STEADY SPEEDS
NO. 19
LEGEND

FIGURE 1.

A ship, in order not to exceed her allowances, must have her plant in prime condition and must be skilled in the operation of the plant.

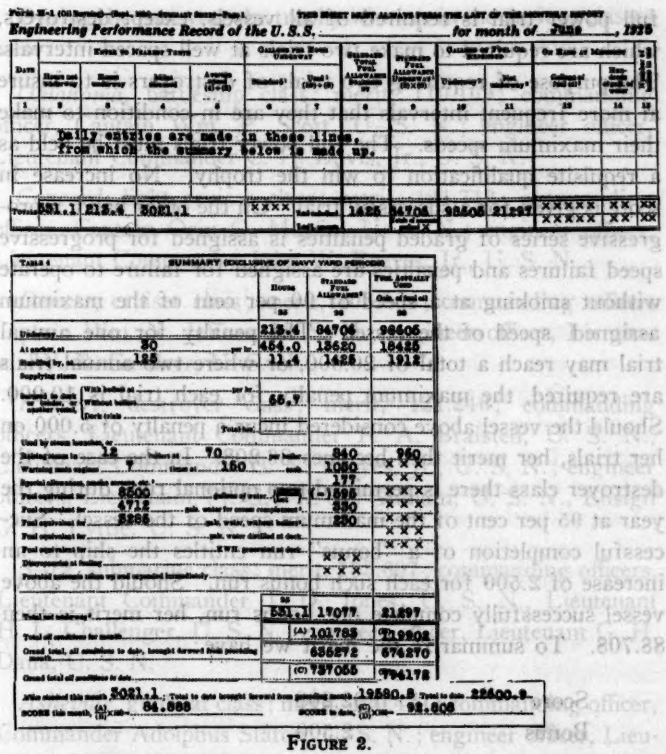
There is shown in Figure 1, a plot of the underway performances of the U. S. S. *Nebraska,* showing fuel consumptions underway for speeds from zero to 19 knots, for the years from 1909 to 1918. The curve of allowances has been drawn through the best performances, and from this curve there is prepared a table of allowances for every tenth of a knot in speed. This method has been in use until the present time and was used in establishing the steaming allowances for 1924–1925. It has been superseded by a more exact method of a percentage of the arithmetical mean of the fuel consumption at each speed as the laying down of the curve shown introduced a personal equation which was not desirable.

The total of the allowances for the month is the total of the allowances for each hour at the speed steamed. The not underway performances are likewise based on assigned standards which are computed from recorded performances. The standing-by allowance in the case of an oil burning vessel is the steaming allowance at zero speed, which represents chiefly the fuel consumed by the auxiliaries in maintaining the vacuum required to get underway and by the fireroom auxiliaries on the additional boilers. In addition to standard allowances there are granted certain additional allowances for warming up the engines, lighting off boilers, carrying extra complement, etc. These are shown in the summary of the report required (Figure 2). The summary furnishes at once a presentation of the ship's performance for ready analysis. The score for the year is determined by the equation:

$$\frac{\text{Fuel allowed}}{\text{Fuel used}} \times 100 = \text{score.}$$

In the case shown this becomes:

$$\frac{737{,}055}{794{,}172} \times 100 = 92.208.$$



Engineering Performance Record of the U. S. S. _____ for month of 1925, 1925

Date	Hours not underway	Hours underway	Miles steamed	Average (Col. 7 ÷ Col. 6)	Gallons per hour underway: Number	Gallons per hour underway: Total (Col. 6 × Col. 7)	Standard Total Fuel Allowance (Col. 6 × Col. 8)	Standard Total Fuel Allowance Underway (Col. 6 × Col. 10)	Gallons of Fuel Oil Received: Underway	Gallons of Fuel Oil Received: Not underway	On Hand at Midnight: Gallons of fuel oil	On Hand at Midnight: Gallons of water
1	2	3	4	5	6	7	8	9	10	11	12	13
	Daily entries are made in these lines, from which the summary below is made up.											
Totals	531.1	213.4	5021.1		XXXX	Total (Col. 7)	1425	84706	98405	21297	XXXXX	XX
											XXXXX	XX

SUMMARY (EXCLUSIVE OF NAVY YARD PERIODS)

	Hours	Standard Fuel Allowance*	Fuel Actually Used (Gals. of fuel oil)
	53	54	55
Underway	213.4	84706	98405
At anchor, at 30 per hr.	464.0	13620	18425
Standing by, at 125	11.4	1425	1912
Supplying steam			
Boilers in use: With bellows at ____ per hr. / Without bellows	66.7		
(Dark trials ____ Total hrs.			
Used by steam launches, at ____			
Raising steam in 12 boilers at 70 per boiler		840	950
Warming main engines 7 times at 150 each		1050	XXX
Special allowances, smoke screens, etc.†		177	XXX
Fuel equivalent for 8500 gallons of water (Evap. 1)		(-) 898	XXX
Fuel equivalent for 4718 gals. water for steam requirements†		330	XXX
Fuel equivalent for 3286 gals. water, without bellows*		230	XXX
Fuel equivalent for ____ gals. water distilled at dock			XXX
Fuel equivalent for ____ KWH. generated at dock			XXX
Discrepancies at trials†		XXX	
Provided automatic stops not operated hospitalably		XXX	
	53		
Total not underway†	531.1	17077	21897
Total all conditions for month		(A) 101783	119902
Grand total, all conditions to date, brought forward from preceding month		656272	674270
Grand total all conditions to date		(C) 737055	794172

Also steamed this month 5021.1; Total to date brought forward from preceding month 19580.9 Total to date 22600.0

SCORE this month, (A)/(B)×100= 94.888 SCORE to date, (C)/(D)×100= 92.208

FIGURE 2.

This score represents the fuel consumption alone. The competition has two other factors, that of reliability and the annual full power trials required. Should a vessel at any time through failure of engineering material be unable to carry out an assigned task, such as failure to get underway at a specified time, or inability to keep place in formation, due to engine breakdown, boiler casualty requiring reduction of speed or a similar casualty, there is assigned a standard derangement penalty of 1.000. Thus the above score of 92.208 would give a merit of 91.208 for one casualty during the year. The annual

full power trial is required of all vessels, except destroyers, which are required to make two trials at well spaced intervals. The purpose of requiring two trials of destroyers is to insure at more frequent intervals that they are in condition to make their maximum speeds. The full power trials must be held as a requisite qualification to win the trophy. No increase in score is given for a successful trial. On the other hand, a progressive series of graded penalties is assigned for progressive speed failures and penalties are assigned for failure to operate without smoking at a speed of 90 per cent of the maximum assigned speed of the vessel. The penalty for one annual trial may reach a total of 20.000, or where two annual trials are required, the maximum penalty for each trial is 10.000. Should the vessel above considered incur a penalty of 5.000 on her trials, her merit then becomes 86.208. In the case of the destroyer class there is permitted two optional runs during the year at 95 per cent of the maximum speed of the vessel. Successful completion of a "bonus" run entitles the ship to an increase of 2.500 for each such bonus run. Should the above vessel successfully complete one bonus run, her merit is then 88.708. To summarize the merit we have:

Score	92.208
Bonus	2.500
	94.708
Less derangement penalty	1.000
Less full power trial penalty	5.000
Final merit	88.708

In addition, it is required that for eligibility the high vessel must have steamed at least 70 per cent of the average mileage of the otherwise eligible vessels, except that in the transport and minesweeper classes this limit is set at 50 per cent.

During the past year the trophy winners in the various classes were:

Oklahoma, battleship class; merit, 110.169; commanding officer, Captain Wm. P. Scott, U. S. N.; engineer officer, Lieutenant Commander C. H. Davis, Jr., U. S. N.

Concord, light cruiser class; merit, 121.716; commanding officer, Captain Orin G. Murfin, U. S. N.; engineer officer, Lieutenant Commander John W. Reeves, Jr., U. S. N.

Denver, cruiser class; merit, 103.986; commanding officer, Captain Wm. N. Jeffers, U. S. N.; engineer officer, Lieutenant W. R. Gardner, U. S. N.

Hulbert, destroyer class; merit, 121.246; commanding officers, Lieutenant Commander F. A. Braisted, U. S. N., Lieutenant Commander G. N. Reeves, Jr., U. S. N.; engineer officers, Lieutenant (J. G.) R. M. Graham, U. S. N., Ensign G. H. Lyttle, U. S. N.

S–12, submarine class; merit, 133.607; commanding officers, Lieutenant Commander J. D. Jones, U. S. N., Lieutenant H. L. Challenger, U. S. N.; engineer officer, Lieutenant G. H. Dana, U. S. N.

Asheville, gunboat class; merit, 120.431; commanding officer, Commander Adolphus Staton, U. S. N.; engineer officer, Lieutenant Robert Anderson, U. S. N.

Camden, tender class; merit, 93.555; commanding officer, Commander W. F. Newton, U. S. N.; engineer officer, Lieutenant W. A. Teasley, U. S. N.

Kanawha, transport class; merit, 106.092; commanding officers, Commander C. E. Smith, U. S. N., Commander E. R. Shipp, U. S. N.; engineer officer, Lieutenant R. Iverson, U. S. N.

Umpqua, mine sweeper class; merit, 108.076; commanding officer, Lieutenant J. C. Rickertts, U. S. N.; engineer officer, Chief Machinist Wade Lash, U. S. N.

The trophy winners in each class are permitted to carry the class trophy for one year. The trophy is a beautiful bronze plaque portraying suitable engineering subjects (see frontispiece). The vessel also paints a large white "E" on her smokepipe, letters of commendation are sent by the Secretary of the Navy to the captain and the officers concerned, cloth "E's" are worn by the enlisted personnel of the engineer department on the uniform, and each enlisted man is awarded a cash prize of $15.00. A ship which attains a merit of 98 per cent of the trophy winner is entitled to carry a red "E" on the smokepipe and to wear red letters on the uniform. Letters of recognition are also sent to the officers. With such splendid inducements the competition is extremely keen. The high scores may be accounted for by the fact that although the allowances are based on best previous information of best performances and designed to allow a score of 100 per cent for the best work, the ships develop ways and means of reducing fuel consumptions that produce results which eclipse all previous performances. The following vessels obtained a final merit of 98 per cent of the final merit of the trophy winner in their respective classes and were awarded the letters and recognition above mentioned:

Richmond, light cruiser class; merit, 121.139; commanding officer, Captain L. A. Cotton, U. S. N.; engineer officer, Lieutenant Commander R. F. Meyers, U. S. N.

Percival, destroyer class; merit, 119.898; commanding officer, Lieutenant Commander C. H. Cobb, U. S. N.; engineer officer, Lieutenant (J. G.) C. H. Router, U. S. N.

R–6, submarine class; merit, 131.965; commanding officer, Lieutenant (J. G.) R. W. Abbott, U. S. N.; engineer officer, Lieutenant (J. G.) R. W. Abbott, U. S. N.

O–3, submarine class; merit, 131.376; commanding officer, Lieutenant C. H. Waters, U. S. N.; engineer officer, Lieutenant (J. G.) R. F. Yager, U. S. N.

S–18, submarine class; merit, 131.258; commanding officer, Lieutenant F. P. Thomas, U. S. N.; engineer officer, Lieutenant (J. G.) W. M. Hainer, U. S. N.

Shawmut, tender class; merit, 92.886; commanding officer, Captain J. W. Greenslade, U. S. N.; engineer officer, Lieutenant T. A. Solberg, U. S. N.

Whippoorwill, mine sweeper class; merit, 106.453; commanding officer, Lieutenant Wm. J. Poland, U. S. N.; engineer officer, Chief Machinist B. McC. Proctor, U. S. N.

In order to afford encouragement to vessels which may have been low and which do not win the trophy, there is awarded in several classes what is known as the Greatest Improvement Prize. This is the same as for a vessel making 98 per cent of the trophy winner except that a money prize of five dollars per man is awarded the engineer's force. During the year 1924–1925 this prize was won by the following vessels:

Nevada, battleship class; improvement factor, 1.196; commanding officer, Captain D. W. Todd, U. S. N.; engineer officer, Lieutenant Commander R. G. Coman, U. S. N.

Henshaw, destroyer class; improvement factor, 1.406; commanding officer, Lieutenant Commander T. F. Caldwell, U. S. N.; engineer officer, Lieutenant (J. G.) V. E. Corns, U. S. N.

O–1, submarine class; improvement factor, 1.246; commanding officer, Lieutenant H. P. Burnett, U. S. N.; engineer officer, Lieutenant (J. G.) A. J. Gray, Jr., U. S. N.

In order to insure that the improvement prize is won through merit alone and not through abject failure the previous year, it is provided that a ship must be eligible both years. It is also provided that the merit for each year used for determining this factor shall not include the full trial penalties, otherwise a vessel which took a full power penalty of 20.000 on the first year might by making a successful trial alone the second year register an improvement of 1.200, assuming the same fuel con-

sumptions for each year with a score of 100 for the first year. Should the score the first year be as low as 50, the factor would then be 1.400. Thus, a vessel which registered a complete failure might receive undeserved laurels. In order to compute this factor, the following formula is used:

$$\text{Improvement factor} = \frac{Y}{Z} \text{ where}$$

$$Y = M \frac{U\frac{Q}{R} + V\frac{S}{T}}{U + V}, \text{ and } Z = \text{first years merit}$$

where:

M = Merit for second year.
Q = First year's allowance underway for average speed made second year.
R = Second year's allowance underway for average speed made second year.
S = First year's "at anchor" allowance.
T = Second year's "at anchor" allowance.
U = Fuel used underway second year.
V = Fuel used not underway second year.

The purpose of the U and V factors is to give proportional weight to the underway and not underway performances. The average speed is used, as the labor involved in using the separate speeds made throughout the year would be prohibitive.

PART II.

ENGINEERING PERFORMANCES OF TROPHY WINNERS.

Considering the keenness of the competition in each class, the engineering methods in vogue on ships winning trophies should merit close study, not alone by naval officers concerned with the operation of naval vessels, but by the engineers in our merchant marine. It is quite possible that some of the infor-

mation given on fuel oil burning could be used to advantage in shore plants. The conditions discussed are, however, particularly those of the Navy, and include a discussion of personnel and military factors. The following discussion is based on information obtained from the ships.

U. S. S. "OKLAHOMA."—BATTLESHIP CLASS.

The winning of the trophy is ascribed to:

(*a*) The satisfactory mechanical condition of all machinery was assured before attempting any methods of analysis, thus eliminating false conclusions.

(*b*) A definite plan of operation was next established, based on actual tests made on each unit.

(*c*) High morale was fostered among the entire personnel.

The mechanical condition of the plant. Particular attention was paid to the tightness of the main engine pistons, pump pistons and plungers, as well as the alignment of all moving parts. The high pressure piston rings were frequently renewed, and intermediate and low pressure piston rings were frequently inspected and aligned. The stuffing boxes were kept tight as well as all joints, all of which materially reduced the water rates and the amount of make-up feed required. Water lost is charged at the rate of .07 gallons of fuel oil per gallon of water and deducted from the allowances so that the saving of make-up feed is a direct gain. A saving of make-up feed also requires less evaporation from the boilers. Attention was paid to the cleanliness of boilers, condensers, feed water heaters, and fuel oil heaters, thus insuring a standard condition of material upon which to base conclusions and judge methods of operation. A series of tests was conducted on the various auxiliaries, establishing the lowest speed at which the auxiliary was capable of accomplishing the assigned task. This low limit of speed and consequent steam consumption is a direct function of the mechanical perfection of the unit. The slower

speeds involved result in a direct reduction of wear and tear, reduced repair periods, and rotation of duplicate units. These facts were impressed upon the personnel.

With material in good condition routine overhaul was reduced, and the time devoted to major overhaul work formerly accomplished by the Navy Yard. This resulted in a direct money saving to the Government in addition to the fuel saving involved, an important item not reflected in the scoring or fuel consumption figures. The efficient maintenance of the plant by the ship's force serves to defer Navy Yard overhaul periods and to assure a higher average material condition throughout the year—an important item in these days when declarations of war are made on short notice.

Operation. The following operating factors were given particular attention:

(*a*) The determination of the proper number of boilers to give both economy and a large amount of reserve power.

(*b*) Especial attention to fuel oil temperatures, air pressures and forced draft blower speeds.

(*c*) Low salinity of boilers.

(*d*) Prevention of steam throttling to main engines by maintaining a pressure in the high pressure steam chest as nearly equal to the line pressure as maneuvering requirements permitted.

(*e*) Large, sudden, and frequent changes of speed were avoided as much as possible. The readjustment of the auxiliary machinery to a new series of pressures and temperatures entailed by a speed change is wasteful because there exists a time factor during which the plant is being readjusted to an economical operating balance. During such periods of readjustment, occurring particularly when speed changes are large, the plant is not operating at its best economy.

(*f*) An inspection system was established providing for critical inspection of machinery operation. Without a system

of follow-up, the established methods of operation will deteriorate, due chiefly to changing personnel or lack of interest on the part of some individual.

Morale. In order to secure the interest and cooperation of the personnel, the following methods were used:

(*a*) Instruction and training.
(*b*) Privileges.
(*c*) Inter-ship competition.
(*d*) Intra-ship competition.

The men were instructed at their operating stations and the relation of their task to the whole plant was constantly kept before them. Engineering information was imparted as rapidly as the men would absorb it and the instruction divested of the "school" character. All officers were utilized for this instruction. The reprinted pamphlets from the Engineering Manual were supplied to the men. Special problems were discussed in conference and brought to the attention of the men where general cooperation rather than specific action was required.

In view of the frequent necessity for repair work outside of working hours, particular consideration was given to the shore leave of the men, involving a departure from the routine. This insured the cheerful accomplishment of work out of routine. A rigid system of discipline was enforced. It is a well established fact that discipline is easily obtained in an organization where a task is being accomplished and where interest is being maintained regardless of the amount of additional or arduous work being done by the men.

Inter-ship Competition. The comparative standings are published monthly by the Division of Fleet Training, office of Chief of Naval Operations, the Engineering Section of which supervises and conducts the engineering competitions of the Navy. These summaries are utilized in presenting the comparative work of the ships to the men both by tables, graphs

and curves. The work of the other ships is usually published in advance within the fleet, although subject to correction, and these reports are followed by the men with the same interest as any baseball fan follows a world series game by innings. Where ships are particularly close, the publication of scores is a spur to renewed efforts.

Intra-ship Competition. Due to the high standing of the ship it was felt that a feeling of security might result in a letdown in performance. To counteract any such tendency, a competition between watches was started which produced remarkably successful results. Scores were computed hourly while underway and sent to the different units of the engineering department for information. The interest thus stimulated was beneficial to the actual going performance of the watch, which is not the case where past information is given out. The officer of the deck was likewise informed of the score made, thus inviting his cooperation. The score for the watch was computed and credit to the officers and men involved duly given on the bulletin board. A monthly summary of these scores was made and published, thus insuring continuous and keen competition between the individual officers and between the units of the men.

The commanding officer of this ship points out the fact that although this ship has been in service ten years, the officers and men do not look upon her as an old ship but as a very good one. During the past year the *Oklahoma* made a full power run without penalty for the first time since commissioning. The vessel rose from fourteenth place in a list of eighteen in 1923–1924 to first place in the same list in 1924–1925. Her improvement factor was 1.294, the highest made, but she was not awarded the Greatest Improvement Prize because of ineligibility in 1923–1924 due to not having steamed 70 per cent of the mileage of the eligible vessels for that year. The mileage requirement is inserted to prevent a vessel which remains at anchor a great portion of the year from winning the trophy against competitors which have to do a great deal of steaming.

The methods in vogue on the *Oklahoma* are sound and worthy of emulation and great credit is due that vessel for her excellent work in reducing her fuel consumptions.

U. S. S. "CONCORD"—LIGHT CRUISER CLASS.

In view of the intense competition between the U. S. S. *Concord* and the U. S. S. *Richmond* which the *Concord* won by a margin of less than six-tenths of a point we may assume that the performances of these two vessels was equally splendid. Between two such performances there is no choice and one must consider them equally efficient insofar as engineering excellence is concerned. The *Richmond's* operating schedule was probably more rigorous. The *Concord* scored 121.716 and the *Richmond* scored 121.139. The *Concord* considers that their small margin of victory was won by refusing to be discouraged and by everlastingly sticking to the job. A gallon of oil saved during a mid watch in port is as good as a gallon of oil saved in a full power run, and the formula is therefore an excellent one worthy of emulation.

This ship made a preliminary survey of all of the factors entering into the competition and as a result of the year's work considers the following factors in the order of their importance as contributing to the successful winning of the trophy.

Leadership. This may be defined as the energy, interest, and enthusiasm of the officer personnel of the engineer department, exerted in such a way as to arouse similar emotions in the men. Leadership must be exerted from the top downward. If lacking at the top the result is no effort at all, or at best, misdirected effort. When present, it produces a sort of momentum which carries the organization along, eliminating the constant driving which is physically impossible to continue indefinitely.

Attention to the upkeep of the plant, especially boilers and auxiliaries. This does not mean that other parts of the plant may be neglected. It does mean that no matter how good other

parts of the plant may be, so long as there are defects in the boilers or auxiliaries the results attained will be indifferent. It is not difficult to figure out the loss of efficiency in a boiler due to dirty tubes, either on the fire side or the water side, nor to figure the loss due to excess air coming in through leaky boiler casings. The size of these losses will probably be surprising. The frequency of their occurrence aboard ship is more surprising. As to auxiliaries, of which pumps are the chief, it is easy to compare their rate of performance with their actual performance as shown by a small measuring tank.

Attention to, and exact knowledge of, the operating efficiency of boilers and of auxiliaries, including traps. This item is similar to the preceding, for it is necessary to have exact knowledge of efficiencies in order to know best when upkeep work should be done. There is required constant use of the gas analysis apparatus, the keeping of the daily analysis sheet, daily test of the operation of traps, and frequent checks on the water rates of pumps. The material supplied for this purpose is not adequate, but, together with such as can be improvised, must be utilized to the utmost. Shore power plants of any size have for years been using automatic recording gas analysis instruments, steam flow meters, automatic recording steam gauges, etc. While some of these instruments have been used to some extent in naval vessels, it is understood that they are not generally supplied because satisfactory types are not available. Yet these instruments appear to be a satisfactory part, and in fact, an essential part, of the equipment at shore plants where conditions are greatly different. It is safe to state that the percentage of increase in operating costs due to the lack of these instruments is very large.

Efficient operation of main generators and of all users of electricity, including an analysis of electrical requirements throughout the ship. The relative importance of this item depends upon the amount of electrical equipment on the ship and the extent of its use. This item is important because of the

ease with which the efficiency of the main generators can be checked up, the efficiency of the electrical units and the fact that when once an electrical machine is performing at its proper rate of efficiency, it will remain in this condition for some time. It is easy to investigate separately the various electrical units throughout the ship, such as baking apparatus, air compressors, winches, etc. The use of more power than is necessary or for a longer period than is necessary can then be readily detected and corrected. This assumes co-ordination between the various departments of the ship.

Efficient operation of the evaporator plant, including an analysis and check on all fresh water users throughout the ship. There will probably always be room for improvement in operation and minor improvements in installation. This presupposes having means of closely figuring the cost of making water. A steam flow meter is almost an essential, but nevertheless almost never available. In addition to analyzing the efficiency of the plant it is necessary to analyze all of the various users of fresh water, including galley, scullery, laundry, washrooms, scuttlebutts, etc. Satisfactory meters are now generally installed, and by their use, the proper expenditure for each user of fresh water can be readily and accurately determined. It remains to see that these estimates are not greatly exceeded. This can be accomplished by constant and intelligent supervision. The locking of washrooms and similar methods are neither necessary nor desirable. There may be included under this head the keeping of records on the quantity of make-up feed required underway and at anchor. A constant look-out must be kept for steam leaks and water drips, and these must be repaired immediately upon discovery regardless of time or place. It will be found that from the number of steam leaks and water leaks observed in a plant the make-up feed rate can be estimated with surprising accuracy.

The active interest and aid of all officers on the ship, especially the Commanding Officer. The part of the Commanding

Officer is merely leadership along broad lines. The subordinate should be able to feel that the interest of the Commanding Officer is present, and that, in making any decision, the Commanding Officer has given his department its share of consideration. With respect to the other officers aboard, active interest and aid is required from them all. No matter who the officer or what his job, if he has no interest outside of that job, he detracts by so much from the team work and co-ordination which must obtain if the ship is to reach her maximum efficiency.

The intelligent assistance of all the crew, particularly the engineer's force. The importance of this factor is not minimized. Actually it seldom attains its proper importance due to such practical considerations as lack of permanency in personnel and shortages in complement, thus reducing the time available for instruction. Instruction and education of the personnel results not only in a better individual performance of duty, but in a better understanding of what is being done and the objectives in view, thus securing more energy and enthusiasm in carrying on the work. This lifts a great part of the load of supervision from the officer personnel. The development of cooperation by the crew is accomplished by the removal of practical difficulties. This cooperation when secured is a powerful factor towards success. The impression persists that people are looking and hoping for some short cut, some easy or new way to success. If so, they will be disappointed. The formula is time worn; hard work and perseverance. Some of the more important factors—leadership and cooperation—concern the ship as a whole rather than the engineer department alone. Here lies the trouble with all written comment on this subject; it is often never seen by officers outside the engineer department, although it concerns them as much as anyone else. The mission of the engineering competition is admirably expressed in the rules, but to be of value it must be known to all of the officers of the ship, and be kept fresh in their minds.

Many lose sight of the mission and many more have never heard of it. The result can be only one thing—loss of perspective and of relative importances. Without perspective the engineering competition is likely to suffer or to receive undue weight in the operation of the ship. It should strike a true course, midway between these two dangers.

U. S. S. "DENVER."—CRUISER CLASS.

The *Denver* won the trophy in the Cruiser Class for the second successive year. During both years this vessel operated under the same Commanding and Engineer Officers. The engineering excellence of the *Denver* is credited to a careful study of the Engineering Manual,* and the Engineering Bulletin† and all aids and suggestions therein applicable to the *Denver* were tried out. Steam lines, pumps and traps were carefully watched for leaks, and immediately corrected where noted. While at anchor, close attention was paid to the cleanliness of fires and maintaining a constant water level in order to insure a steady rate of steaming. At sea, each fireroom watch was placed on a fuel allowance for a given speed and the performance of the watches compared. No water was purchased by the ship during the year, as the new low pressure evaporators supplied all ship's needs without difficulty. The amount of fresh water allowed was liberal and was authorized with the proviso that the crew would cooperate by turning out unnecessary lights and securing unnecessary machinery. This ship makes the point that the trophy was won without discomfort of any kind to the crew.

* The Engineering Manual is a publication issued by the Bureau of Engineering containing separate chapters for each type of machinery, both electrical and mechanical, installed on board Naval vessels. Its pages present in minute detail the instructions for the care, preservation, repair and operation of practically all marine units. This Manual is the composite publication of the best operating engineering information in the Navy and the author considers it the finest publication of its kind ever published. Its binding permits of changes and additions which are frequently issued.

† The Engineering Bulletin is issued by the Bureau of Engineering from time to time. It contains announcements of developments, special repair methods developed by ships and short descriptive articles of current engineering interest.

U. S. S. "HULBERT."—DESTROYER CLASS.

This vessel has won the trophy for two successive years. Considering that there are 108 vessels in competition in this class, this is a remarkable result. The methods in use on the *Hulbert* stress an intensive supervision of the material condition of the separate units. Particular care is given to the boiler bricking and the maintenance of air tight casings, together with maintaining a water salinity in the boilers below 5 of chlorine at all times. This is very unusual and difficult of attainment, and is accomplished by washing out boilers between overhauls. The condition of the boiler tubes after 1000 hours of steaming and the overall efficiency of the plant compensates for the loss of fresh water entailed.

The main turbines are kept in condition by overhauling and cleaning all bearings, stoning journals, and thrust collars and checking thrust shoe surfaces in a surface plate. The main condensers are kept clean by boiling out, and the lubricating oil is renovated about every two weeks where the vessel has been steaming. Bearings of forced draft blowers are checked up at least once a year, and pumps periodically checked up and overhauled.

While underway the back pressure is utilized for feed heaters and any excess put into the hot well direct; whereas the evaporators are shut down as much as possible at sea to favor this production of hot feed water. This method has increased the hot well temperatures 30 degrees, the water reaching a temperature of 160 degrees in the hot well. The make-up feed is reduced to 7.2 gallons per mile, which is very low. This vessel, of course, uses Borneo oil to a great extent, and this oil, being very light, permits burning at temperatures from 90 degrees to 110 degrees F., whereas the average West Coast oil must be heated to around 170 degrees. Lower air pressures are also required for atomization.

For port operation, boiler pressure is reduced to about 105 pounds gauge. This is possible because no auxiliary requires

more than 80 pounds pressure. This results from tight auxiliaries and the low boiler pressure in turn causes less loss from radiation and from leakage as some leaks do not open at the lower pressures.

The personnel is instructed in much the same manner as on the *Oklahoma* and the morale is high. The reports from ships doing excellent work invariably report a high morale. The men are examined for promotion and advanced whenever possible.

U. S. S. "PEARY."

This destroyer was not one of the trophy winners, but from 1921–1924 has reduced the fuel consumption at anchor by such a large amount as to cause the Department to inquire into the methods used. The fuel consumptions of the *Peary* will be shown in percentages based upon an average fuel consumption of 100 per cent in 1921–1922 by the 19 vessels in the Cramp-Parsons group. All percentages shown are percentages of this year's average.

Year	Average fuel used at anchor by 19 vessels in the group.	U. S. S. *Peary*, fuel used at anchor.
1921–1922	100.000	91.837
1922–1923	76.531	61.224
1923–1924	76.275	73.980
1924–1925	83.418	55.296

The ship ascribes the low fuel consumptions to permanency of personnel, and a personal interest by the men which made the difference between ordinary and efficient operation. The operation of the evaporating plant is credited with the major portion of the reduction in fuel consumption. Although run on back pressure exclusively, there is a decided increase in fuel consumption when the evaporators are running, sometimes as high as 200 gallons of oil a day. Therefore, the more water made and the more water saved, the longer the evaporators may remain secured. When they are running the generator, it is run against back pressure, the auxiliary condenser secured

and all drains cut in on the hot well. There is a small loss of water due to steam exhausting to the atmosphere through the hot well. In the evaporators, number two oil cooler pump pulls 18 inches of vacuum on the shell. Due to the size of the pump, priming will occur if the shell is more than half full, or if a higher vacuum is carried. A fire and bilge pump is used as a brine pump taking suction from the bottom of the shell and from the fire main, and discharges to the shell through the distillers and feed water heater, and overboard. The water of the distillers is kept no higher than luke warm. A better vacuum can be pulled by use of the evaporator feed pump, but there is no gain in output, so this pump is put on the fire main and furnishes an ample supply of flushing water at all times. Using this hookup, 1000 gallons a watch can be made on ten pounds of back pressure and the coils do not need scaling oftener than twice a year. An additional advantage of the brine pump is that the evaporators require a bottom blow only every forty-eight hours. When the evaporators are running, 120 pounds of boiler steam is carried; when secured, 90 pounds is carried. Eighty pounds can be carried but the fuel oil service pumps, which require the highest steam pressure, do not run smoothly due to excessive condensation. Fuel oil pressure is maintained at 150 pounds and the oil heated to about 200 degrees F., depending upon the oil. The feed water temperature is carried as high above 220 degrees F. as possible. By cooperation from the deck force in the matter of deck pumps, anchor engine, steam to showers, etc., the use of extra power is kept to a minimum. A steam line is connected to the galley from number two fireroom so that when the fireroom is in operation, all steam lines may be secured in number one fireroom.

The main items in economical operation may be summarized in order of importance as follows:

(*a*) Fostering the spirit of competition.

(*b*) Securing maximum capacity of evaporators.

(*c*) Keep boilers clean (clean every 300 hours).
(*d*) Overhaul all auxiliaries frequently.
(*e*) Stop all leaks—internal and external.
(*f*) Determine most economical pressures and temperatures and maintain them.
(*g*) Cut down demand for extra power to a minimum.
(*h*) Keep two men on watch in the fireroom at all times.

Such means of obtaining false economy as securing a generator in the daytime are not practicable. There was no curtailment of water, light, heat, showers, or ventilation.

U. S. S. "S-12."—SUBMARINE CLASS.

Vessels of the Submarine Class have only during the last two years had their comparative performances placed on a fuel per mile basis. Trials are also required as heretofore, but do not form the sole basis of competition. The fuel per mile figure for a Diesel engine under similar conditions is more or less of a constant at all speeds and the reduction of this figure represents the overall improvement of the condition of the engines. The use of a standard figure based upon attainable results permits the individual ship figure to become the index of the efficiency of the engines of that ship. While this is the major factor of submarine performance as relates to fuel consumption, it is only a part of the competition for these vessels. The efficiency of the electric installation enters for submerged operations. In addition, two full power trials are required annually, and two availability trials are required quarterly to insure a continuous state of readiness throughout the year. The full power trial consists of a long endurance run on the surface, a shorter submerged endurance run, a full power run on the surface, and a shorter full power submerged run. Between these trials and as a continuous part of the whole, there are required quick dives and battery and air bank charging operations which are performed under the specifications of the rules and involve penalties for failure. The availability trials are

an abridged form of the full power trials. Penalties are likewise involved for failure to make availability trials, but a bonus is awarded for successful performance, which encourages their successful completion. Derangement penalties for unreliability during normal operations are assigned as in the case of surface craft.

The performance of the trophy winner in this class is worthy of close study by competing vessels. During the competition year 1924–1925, the U. S. S. *S–12* was operated whenever possible at the speed which was most economical.

A distinct aid to economy and reliability of the engineering plant was the method of station keeping in vogue for long trips. During the winter cruise submarines were required to be at a certain interval from the tender and to be in a certain sector, generally 45 degrees. This allowed the boats to run for hours without changing throttle openings. Formerly, when required to keep exact distance as well as interval it was the usual thing for submarines to change the throttle ten to twenty times a watch. A submarine, if required to do so, can keep as accurate position as any other type of vessel, but to do so produces considerably more wear and tear on the material on this type of vessel than on other types of vessels. For best results a float should be carried on long trips for about two-thirds of the time and when carrying a float a change of even four turns requires a mutual adjustment of the throttle and fixed current by the throttle-man and electrician. This takes the throttle-man away from his more important duties—watching his engines' pressures and temperatures. Further it is an undisputed fact that a Diesel engine will run more economically and with less wear on parts if speed and power are constant. The standard speed of the Submarine Divisions on long trips was the cruising speed of the submarine longest out of dock. This prevented the overloading of a boat with foul bottom if a boat with a clean bottom was guide. Eight months difference in docking means sixty turns difference for same speed over the ground.

While cruising at speeds of about 300 R.P.M. the battery was floated on the line at a rate sufficient to care for the auxiliary load and also bring the battery to a higher state of charge. The efficiency of a Diesel engine is of course higher, the greater the load up to about full load. The higher propeller efficiency and higher engine efficiency resulted in considerable saving of oil.

The allowance of oil for charging batteries is sufficient when the engine is charging at rates above 1000 ampères with batteries in series but at the finishing rate the fuel used was considerable more than that allowed. The rules for 1925–1926 provide for "overhaul condition" periods, during which time, fuel used for charging batteries will not affect the score of the vessel. This should be helpful to the condition of the battery as it will enable a Commanding Officer to put in more equalizing charges without adversely affecting the score as it did under the old rules. The allowances for averaged overall charging rates were, however, equitable.

It was noted a few days after leaving the Navy Yard that when safety tanks are blown an oil film was present on the water. This became greater from time to time and also developed in No. 2 main ballast tank. Finally oil became noticeable in the wake of the vessel. To prevent this loss which, though at a very low rate, was becoming serious, another method of using the compensating system was adopted. The following method is admittedly artificial but is worthy of consideration in view of results obtained. Instead of the man on watch in the engine room forcing oil to the gravity tanks by the circulating water pressure or by sea pressure, whenever the gravity tank needed it, *once* a day the oil was forced back to No. 7 or No. 8 tank until it was full and oil was then *pumped* from No. 7 or No. 8 fuel tank to the gravity tank as required. This method kept pressure off the fuel system except for about 40 minutes a day and also enabled an accurate fuel account to be kept because No. 7 or No. 8 fuel tank can be sounded.

Results showed that during May and June over one thousand gallons of fuel were saved; this amounted to about 1.5 gallons per hour underway.

In peace times it is almost always known when a boat is expected to dive, so that the compensating system can be cut in in sufficient time in order to have the boat in proper trim. This will not slow up the dive more than a few seconds and an adjustment of trim can be made very shortly after submerging. The present fuel tanks in submarines do not remain tight with the present method of compensating. The tanks are tight when tested in the Navy Yard but after one dive and one engine run they begin to leak due to strains and vibration. Fuel compensating as generally used is too costly for peace time operations.

As a matter of further supervision, while cruising, the petty officer in charge of each watch reported the amount of fuel used and average turns to the Commanding Officer. Any unusual expenditures were carefully checked up. For the various speeds the proper throttle openings were known and if the throttles were not on the proper notch or both throttles not on corresponding notches, investigation was made and the matter remedied. This applied also to air compressor openings.

In port advantage was taken of overhaul periods, and a very high percentage of the work was in the nature of preventative overhaul. Before arriving in port indicator cards were taken at high speeds, if possible, in order to check up work to be done and parts to be inspected. Indicator cards were taken frequently on long cruises. Indicator cards were also taken in port just before long cruises. These cards were taken after the engine had been accurately timed and timing again checked if necessary. Especial care was taken to keep fuel clean, also to keep spare valves clean and to keep the fuel pump tight.

During long cruises the battery was kept at about half capacity by "floating" on line so that if it became necessary to shut down one engine for inspection or minor repairs the

vessel, by running on one engine and one motor, could keep position. Exhaust valves have been renewed, air cooler tubes plugged and evaporators cleaned out without losing position. The time necessary to do any of the above three operations was from one to one and one-half hours. The above repairs were made very rapidly due to the excellent team work of the engineer's force and to the fact that all possible tools and material were laid out and ready for use before the engine was stopped.

If the overhaul periods had not been provided and strictly adhered to in accordance to schedule, the submarines would not have proven reliable during the winter cruise. Upon arrival in port after a long trip a certain number of days overhaul period was assigned during which time boats were *not allowed* to operate.

Important as is the question of personnel considerations in its relation to engineering performances on surface vessels, it is even more so on submarines. The unnatural conditions of living are reflected in the physical condition and spirit of the men, and, if their welfare is not carefully considered, the performance of the vessel must reflect the lowered physical and psychological condition of the men. The *S–12* considers that 50 per cent of the personnel contentment factor on a submarine may be charged to food and therefore paid particular attention to this matter. Pride of performance minimized abnormal living conditions. The necessary interest was carefully fostered by official supervision and covered the following points. Comparative standings were published monthly. The engine room was made a show place, battleship style, to such an extent that officers of the Submarine Force and their friends visited it, praised it, and left pleased. This pleased the crew, which extended its efforts to keeping, watch by watch, a spotless, shiny engine room beyond legitimate demands. Inasmuch as this vessel won the trophy, it cannot be said to have detracted from necessary repair work. It served its purpose as a hobby.

A close study was made of derangements occurring on other

vessels and preventative inspections and repairs made to prevent their occurrence on the *S–12*. Times required to effect standard repairs by other vessels were noted and quietly improved upon. This vessel makes the point that full advantage was taken of all overhaul periods so that the vessel went to sea prepared for any operation required of it. Full advantage was taken of the bonus provisions of the rules and one hundred per cent of the possible allowed bonus points were made. Only one other submarine did this.

U. S. S. "ASHEVILLE."—GUNBOAT CLASS.

This vessel has won the engineering trophy in the Gunboat Class for three successive years. The vessel was assigned the full penalty of 20.000 for exceeding the limiting turbine relief valve settings on the full power trials for 1924–1925, but her lead was so great that she won the trophy nevertheless. This vessel was first commissioned as a coal-burning vessel in 1920, was converted to burn both coal and oil in 1921, and in 1922 to fuel oil alone. The improvements made by the operating personnel are responsible for the unusual performance of the vessel. The improvement has been progressive from year to year, so much so that the ship in each case bettered the progressively reduced standards or allowances.

In the engine room, the exhaust from all machinery except the turbo-generators was piped to the back pressure exhaust line. A branch was installed to permit the exhaust being cut into the low pressure turbine while underway. Another connection was made to permit the excess auxiliary exhaust being cut into the evaporators, thus permitting the evaporators to be operated on low pressure exhaust steam. The ship's force constructed a vacuum pump which gave a 15-inch vacuum on the distiller condenser, thus permitting a larger temperature range in the distillation process. A drain feed heater was constructed which utilized the coil drain heat for heating the salt water feed. The evaporator heads were lagged with asbestos and

metal sheeting to reduce heat radiation. White metal plungers were used to replace the packed pistons in the forced lubrication pumps, as it was found that the flax packing disintegrated and found its way into the oil system. The lubricating oil temperatures were reduced by changing the flow of the circulating water in the coolers. The vacuum on the main turbines was increased from 25 inches to 29 inches at full power by the replacement of the heavy springs in the main air pump with lighter ones and by a thorough elimination of leaks in the vacuum system. Due to the height of the sanitary pumps above the injection, it was found that these pumps would not take suction as the auxiliary air and circulating pump took the entire flow, due to their lower level. This required the operation of an uneconomical fire and bilge pump. To overcome this difficulty a sea stool was installed at first dry docking, at a level higher than the sanitary pumps. By an additional cross connection installed for the purpose, the ship was enabled to use one pump for the distilling condensers, ice machines, and sanitary system. The fire and bilge pumps were modified so as to change the water chamber to produce smoother operation. Extensive lagging of pipes was undertaken.

In the firerooms soot cleaning doors were installed on the side casings of all boilers for the purpose of inspection and more thorough cleaning of tubes. The boilers were run down and cleaned after every run, and twice a year the boilers were boiled out with kerosene, boiler compound and caustic soda. An electric motor was used to drive a frayed wire cable for cleaning the tube ends where clay and other foreign matter lodged. A cross connection between the auxiliary feed pump discharge and the engine room feed heater was installed to obtain hot feed water when the auxiliary feed pump was used, thus permitting the main feed pump to be secured for overhaul without loss of economy.

The ship attributes the greatest economy to the method of burning fuel oil, and this statement is equally true for any vessel. It is a fact that for normal demands an oil burning installation will always easily produce the required steam, and the absence of the difficulty of maintaining steam, which difficulty is so common to coal burning operation, serves to induce a false sense of satisfaction which takes no account of economy and only becomes critical upon maximum full power demand. On the *Asheville* the pressures on the fuel oil system were gradually reduced until further reduction would result in lowering steam pressures below the required amount. The booster pumps were thus able to furnish the required pressures even on full power trials. Pressures as low as 60 to 70 pounds with 1/16-inch tips were used in port. For full power runs the writer sees no merit in securing the service pumps, as the possible hazard of pump failure is not warranted by the small economy obtained for the short period involved. The method of reducing speeds to the lowest possible limits and then stepping up the speeds a small amount is commended as a good economy rule to use for the operation of all auxiliaries.

The ship found that the five large ventilating sets, circulating steam heated air, were uneconomical, and built steam radiators which were used instead, so that only one ventilating set was used.

The work of the *Asheville* is noteworthy for the marked originality and resourcefulness shown. The changes made were obviously the result of a thorough knowledge of the plant and its possibilities for improvement. The size of the vessel's installations being smaller throughout permitted the ship's force to make important changes which might be prohibitive on larger installations.

U. S. S. "CAMDEN."—TENDER CLASS.

This vessel competed with tenders and repair ships in the Tender Class, commencing on October 1, 1924. This is the

first time these vessels have been in competition. During port operation a vessel acting as a tender or repair ship uses more auxiliary machinery than vessels of any other type, and consequently has a large amount of auxiliary exhaust. The exhaust is more than is required for feed water heating and to be condensed requires the use of additional steam for the auxiliary condenser. The specific problem involved was to utilize the heat from auxiliary exhaust in the evaporators. The evaporators installed were of the high pressure type, and additional piping was made and installed by the ship's force to permit the evaporators to be operated as a low pressure plant. Where this conversion is made it becomes necessary to increase the vacuum on the distillers in order to increase the range of heat extraction between the low pressure steam supplied and the discharge temperatures of the last evaporation. A small duplex feed pump was installed as an air pump and a vacuum of 10 to 12 inches was secured. Operating on live steam, an output of 14 to 16 thousand gallons of water was obtained with a direct additional fuel expenditure. By utilizing the exhaust steam without a direct fuel expenditure, from 8 to 9 thousand gallons of water was obtained.

In the fireroom careful attention was paid to routine boiler cleaning, the fireroom personnel were instructed in the proper methods of burning coal and a close watch kept on the fuel expenditures. Radiation losses were reduced by lagging, and loss of feed water reduced by correcting leaks. The personnel was given weekly talks on fuel and water economy, thus maintaining interest in operation and economy.

Considering that tenders and repair vessels have facilities at hand for major repair work, these vessels should be in a position to improve the operation of the plants by a considerable amount. The *Camden* is a German built coal-burning vessel.

U. S. S. "UMPQUA."—MINE SWEEPER CLASS.

The *Umpqua* attributes the winning of the trophy to the fact that a regular inspection of machinery was carried out and

that boilers were systematically and regularly cleaned. Due to the nature of the work, steady steaming was possible, which assisted in the saving of fuel. Tubes were blown daily at sea. Feed temperatures were maintained at 230 degrees F. and from 65 to 75 pounds of oil pressure carried. The fuel oil was heated to 250 degrees F., which appears to be very high, and which at full power might serve to decrease the weight of oil delivered. The steam and exhaust piping has been changed to permit the duplex air pump to operate each steam cylinder independently, a very novel arrangement. This arrangement gave 26 inches of vacuum at 11½ knots. The engine room bilges were allowed to accumulate water before using the pumps which were secured upon completion of the operation. In addition, the ship maintained a constant watch for steam leaks. The men were interested in the score and a system of weekly score averages by watches was instituted, with an extra shore leave for the high section.

This vessel is one of the 1000-ton ocean going tugs constructed during the late war period. It entered a new competition class, the Mine Sweeper Class, on October 1, 1924, at which time all auxiliary vessels were assigned to one of three competition classes. The Mine Sweeper Class also includes 34 vessels of the Bird Class in addition to the tugs. The allowances for all ocean going tugs of the 1000-ton class were made the same; likewise the allowances for all vessels of the Bird Class were the same. The fact that a tug stood first and a mine sweeper second would indicate that the allowances were equally correct and that the best vessel won. The operation of the engineering plant to a set of standards and for scored economy was a new departure for these vessels, which up to the present had been operating with only one object in view, that of ability to move. The fuel consumptions for sister vessels differ so greatly that it is at once apparent that many of them have been running with everything wide open and that economy has had no place in the plants' routine. The leading vessels in both the

tug group and Bird Class group produced results in fuel consumptions which indicated that those concerned in the operation of the plants had been previously maintaining the material in a high state of efficiency and had applied the principles of economical operation to their plants during the nine months' period of the competition. Probably no part of the work of the engineering competition stands out in clearer justification of the competition than the widely varying results of its application to a virgin field. Fuel consumption figures produced by the leaders will be improved by them and approached or bettered by those now at the bottom, after these vessels have been in competition for several years. We will thus arrive at a standard and more uniform performance which will result in a large reduction in fuel consumption and repair work.

SIX YEARS WITH THE NAVY IN HELIUM PRODUCTION.

By Lieut.-Comdr. Z. W. Wicks, U. S. Navy, Member.

The successful solution of the problem of the extraction of Helium from Natural Gas on a commercial scale, and at a cost sufficiently low to permit of general commercial use, is one of the greatest engineering achievements of the present century. The construction and operation of the only Helium Plant in the world was supervised by the Navy for a period of about six years, two of which were devoted to construction work, three to operation and one in an inoperative status due to lack of funds. Yet in these three years of actual operation, the production of Helium has been brought from a laboratory curiosity, costing about $1500 a cubic foot, to a commercial product, costing less than two and a half cents a cubic foot.

The production of Helium on a commercial scale was occasioned by the war-time demand for a non-explosive gas suitable for the lifting medium of lighter-than-air ships for military purposes. Since then, there has developed a commercial demand for the same purpose. The Navy Department, working with the Bureau of Mines, has demonstrated the value of Helium to replace the nitrogen in normal air used for divers and tunnel workers. So, from a war-time necessity, comes a product with a rich humanitarian value. The Bureau of Engineering is considering the possibilities of Helium filled searchlights, and the Electrical Industry cannot neglect the advantages of Helium for the cooling medium of large generators and motors.

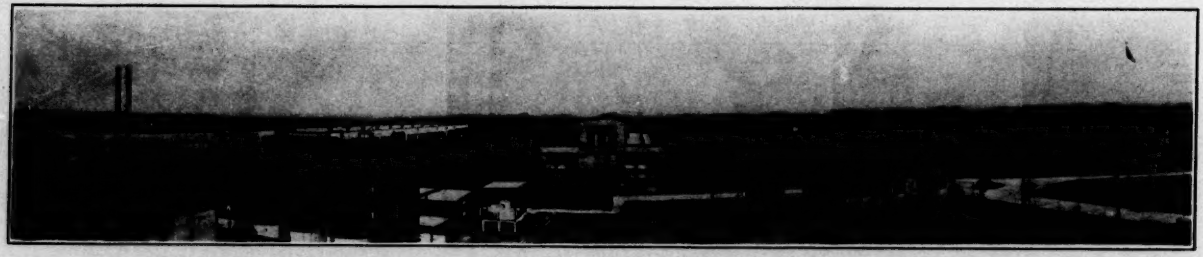
The experimental work in connection with the extraction of Helium from Natural Gas was commenced by Prof. J. C. McLennan of the University of Toronto in 1916, working with the Natural Gas of Canada containing only one-third of one per cent Helium. Shortly after the United States entered the World War, in 1917, the allied military authorities brought the Helium problem to our attention. This led to the establishment of three Experimental Plants in Texas. It was through the personal efforts of Rear Admiral Robert S. Griffin, who was Chief of the Bureau of Engineering, and Engineer-in-Chief of the Navy in 1917, that the Linde Air Products Company was induced to work on the problem of Helium extraction along with the Air Reduction Company and the Bureau of Mines with the Jeffries-Norton system. The proof of Admiral Griffin's judgment lies in the fact that the Linde Company developed the only successful process for the extraction of Helium at that time. The experimental plant built and operated by that Company for the Government was put on a production basis in September, 1918, and produced about two hundred thousand (200,000) cubic feet of Helium, which was on the docks at New Orleans ready for shipment to Europe when the Armistice was signed.

It was about this time that the joint Army and Navy Helium Board came into control of the Helium activities of the Government. This Board was composed of a member, an associate member, and a civilian engineer for each the Navy Department and the War Department, who were the direct representatives of the respective Secretaries. On July 1, 1925, upon the transfer of the Helium activities to the Bureau of Mines in accordance with an Act of Congress, the members were: Captain E. S. Land (C. C.), U. S. Navy, Chairman; Lieutenant Colonel Ira F. Fravel (A. S.), U. S. Army; Lieutenant Commander A. H. Gray, U. S. Navy; Captain R. E. O'Neill (A. S.), U. S. Army; Mr. C. E. Earle, Navy Department; Mr. George Erlandson, War Department.

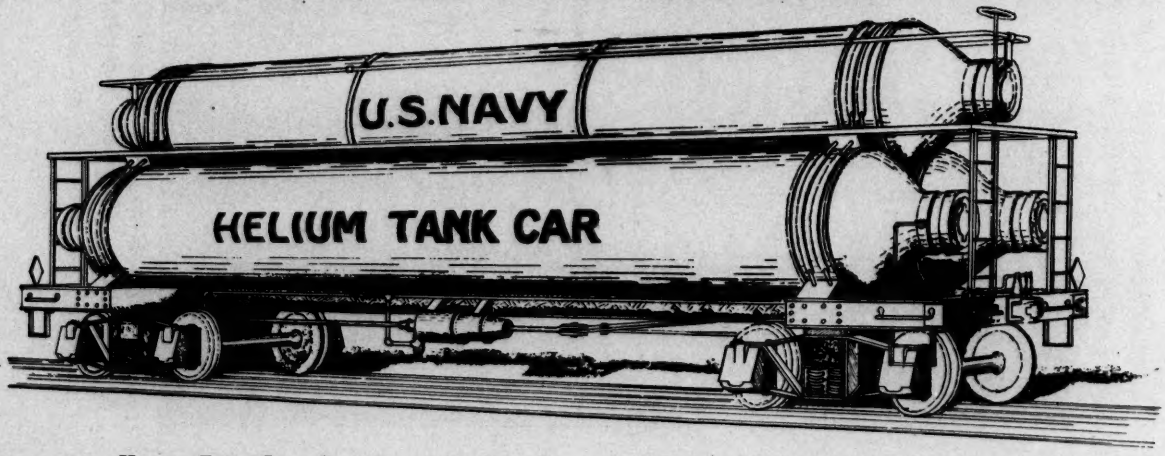
After the end of the war, when it was decided to continue the work for the benefit of the Army and the Navy, the Navy Department was designated as the operating agent of the Army and Navy Helium Board. All matters of policy and distribution of Helium were settled by this Board, but the actual work, and the solution of the engineering problems involved were handled by the Navy Department, at first by the Bureau of Engineering under Rear Admiral R. S. Griffin and after September 1, 1921, by the Bureau of Aeronautics under Rear Admiral W. A. Moffett.

The driving through of the design and erection of the U. S. Helium Production Plant at Fort Worth, Texas, fell to the lot of the Navy Department. A construction contract was entered into with the Linde Air Products Company, which at that time held the only commercially successful process for the extraction of Helium from natural gas, for the construction and erection of the special equipment necessary to process five million cubic feet of natural gas a day and to produce about forty thousand cubic feet of Helium therefrom. The design and erection of the necessary buildings, the purchase and erection of all other machinery and equipment were handled by contracts under supervision of the Bureau of Engineering and the Bureau of Yards and Docks. Contracts were let and work started in May, 1919, and the Plant was finished and placed in operation in April, 1921, when the two year construction period ended.

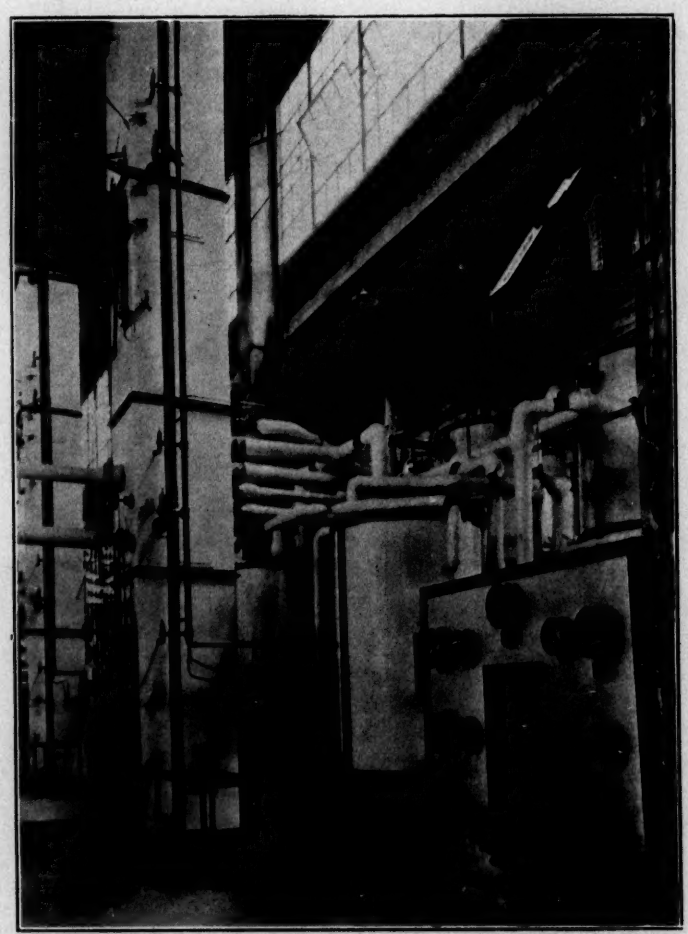
In order to benefit from the experience which the Linde Company had obtained in the production of oxygen from the liquefaction of air, and in the experimental helium plant operated by that Company, a contract was entered into by the Navy Department with the Linde Company, under which that Company would operate the special liquefaction apparatus and the other equipment directly connected therewith. The rest of the facilities of the Plant were directly under the Naval Officers attached to the Plant, who were also responsible for the general control of the whole Plant.



Panorama Showing the Helium Plant, Fort Worth, Texas.



Helium Tank Car. Capacity 200,000 Cubic Feet at 2000 Pounds Per Square Inch at 70 Degrees F.



Interior of Separation Building, Showing Separators in Which Helium is Extracted From the Natural Gas.

The Plant commenced operation in April, 1921, and continued to produce steadily until November of the same year, when it had to shut down due to lack of funds. On account of the inherent difficulties experienced in starting up a new plant, and especially the first one ever built, the cost of production was high, about $500.00 a thousand cubic feet for May, 1921. By November of that year, when the Plant was shut down, the cost had been reduced to $135.00 a thousand cubic feet.

Congress, at the next session, having made ample provision for the Helium Plant, operation was again commenced in October, 1922, with an initial cost of $243.00 per thousand cubic feet, which was reduced to $114.00 by December. During 1923 only sufficient natural gas was available to run the Plant at about half capacity, and the cost was gradually reduced to $82.00. In December of that same year a little more gas was available, and with this increased production the cost fell to $74.00. For the first month since the Plant had been in operation it was able to obtain sufficient gas to operate at full capacity during January, 1924, when the cost was reduced to $65.00.

Now followed the period of improving the process, as it was believed that the lowest cost had been reached for the process as originally installed. By June, 1924, sufficient progress had been made so that with sufficient gas available to operate at four-fifths capacity, the cost was reduced to $61.00. During the summer a high pressure carbon dioxide removal system was developed, designed and constructed by the Naval Officers attached to the Helium Plant. This system was placed in operation on the thirtieth of October, 1924, and although there was only sufficient gas to operate at three-fifths capacity during November, the cost was reduced to $55.00.

It is necessary to remove the carbon dioxide from the natural gas by a chemical process before the gas is introduced into the liquefaction apparatus because at the low temperatures

necessary to liquefy the natural gas the carbon dioxide contained therein is solidified and insulates the apparatus and finally plugs up the small tubes so that the unit has to be shut down and thawed out. The removal system as originally installed was what is known as a low pressure milk of lime method. The natural gas entering the Plant at pipe-line pressure, of about 200 pounds per square inch, was reduced to atmospheric pressure in two pressure reducing valves. This low pressure gas was then led into a nest of towers, thirteen feet in diameter and twenty-five feet high, in such a manner that the gas was scrubbed by the milk of lime solution which was sprayed into the top of the towers. In this process, ten 50 H. P. motor driven pumps were required to circulate sufficient liquid to scrub five million cubic feet of gas a day. The carbon dioxide content was reduced from 0.15 per cent to about 0.05 per cent by this method which, of course, was of material assistance, but not sufficient to accomplish what was desired. Under this condition a separation unit was able to operate about eight or ten hours before slugging up. The low pressure gas was then compressed in large four-stage compressors to about 2500 to 3000 pounds per square inch.

In order to increase the productive life of the separation units and to take advantage of the pipe line pressure, the high pressure caustic soda removal system was developed. Every effort was made to obtain data on such an apparatus, from the various industries of the country, but none was located. The system as developed by the writer and ably assisted by Lieutenant R. E. Davenport, Engineer Officer of the Plant, consists in obtaining an intimate mixture between caustic soda solution and the natural gas at pipe-line pressure in order to obtain the necessary chemical reaction. A special eight-stage, 700 foot head, 3600 R. P. M., centrifugal pump is used to introduce the solution in the gas. On account of the serious action of caustic soda on soldered joints, it was paramount that every trace of caustic soda be removed from the gas after

the carbon dioxide was removed. It could have been scrubbed out with water at a large expense. The problem was solved by Lieutenant Davenport developing a special moisture and vapor trap which successfully removes all traces of the caustic soda. One 60 H. P. motor driven pump is capable of circulating enough solution to remove the carbon dioxide from twelve million cubic feet of gas a day as compared to 500 H. P. required for five million cubic feet with the old process. In addition to this power saving, the first and second stages of the compressors were by-passed and the third and fourth stages used to boost the pipe-line pressure to the required pressure. By a suitable arrangement of piping, the first and second stages of the high pressure compressors are utilized to compress the processed gas in order to return it to the Gas Company. This arrangement, while not recommended for a new installation, is suggestive of making the most use out of what is available when funds for new equipment are not at hand. The total effect of the above changes was to cut the total power requirement of the Plant by more than twenty per cent. In addition to this, the carbon dioxide content of the gas is reduced to about 0.002 per cent and the life of a separator, before slugging up, is now between ten and twelve days instead of between eight and ten hours.

Due to the high efficiency (98 per cent removal) of this new system, the Linde Company was able to try out a major modification to its liquefaction system. Work was commenced on this during the latter part of November, 1924, and by the first of the new year the alterations were completed. This change did away with the auxiliary nitrogen refrigeration cycle, trebled the capacity of each separation unit and increased the recovery of Helium from the natural gas by nearly forty per cent, or in other words, the percentage recovery was raised from about fifty per cent to better than seventy per cent. The combination of this change and the carbon dioxide cycle resulted in reducing the power requirement to

only one-fifth of what it was in January, 1924, for processing the same quantity of gas. In spite of the fact that the extreme shortage of gas in January, 1925, permitted operating at only one-third capacity, and the operators were not entirely familiar with the new process, the cost for the month was down to $51.00.

All of the large gas compressors and the rest of the equipment of the Helium Plant are driven by electric motors. From the time the Plant commenced operating in April, 1921, until March, 1924, all of the power consumed was purchased from the local commercial power company. The Helium Board approved the recommendation of the Bureau of Aeronautics that natural gas engine driven electrical generators be provided to furnish power for the Plant. During the fiscal year 1923, three 500 H. P. units were contracted for and their installation was completed in March, 1924. The purchase of these units was made possible out of savings of the current operating appropriation. Further reduction in the operating costs in the next fiscal year made it possible to purchase four additional units. At the time the Power Plant was planned, the power requirements would have necessitated the installation of sixteen units of 500 H. P. each, but by the time the installation of the seven units was completed in January, 1925, these seven were capable of producing sufficient power for processing up to six million cubic feet of gas a day. This power plant uses, for fuel, the natural gas after the Helium has been extracted. An additional saving is effected in that all of the gas consumed in the engines does not have to be recompressed for delivery to the Gas Company. The average cost of purchased power was about twelve (12) mills per K. W. H., whereas it is produced at the Helium Plant for from five to five and a half mills.

With this saving in the cost of power and the operators becoming more familiar with the new process, the cost of production was driven down to $32.00 in March, 1925. This

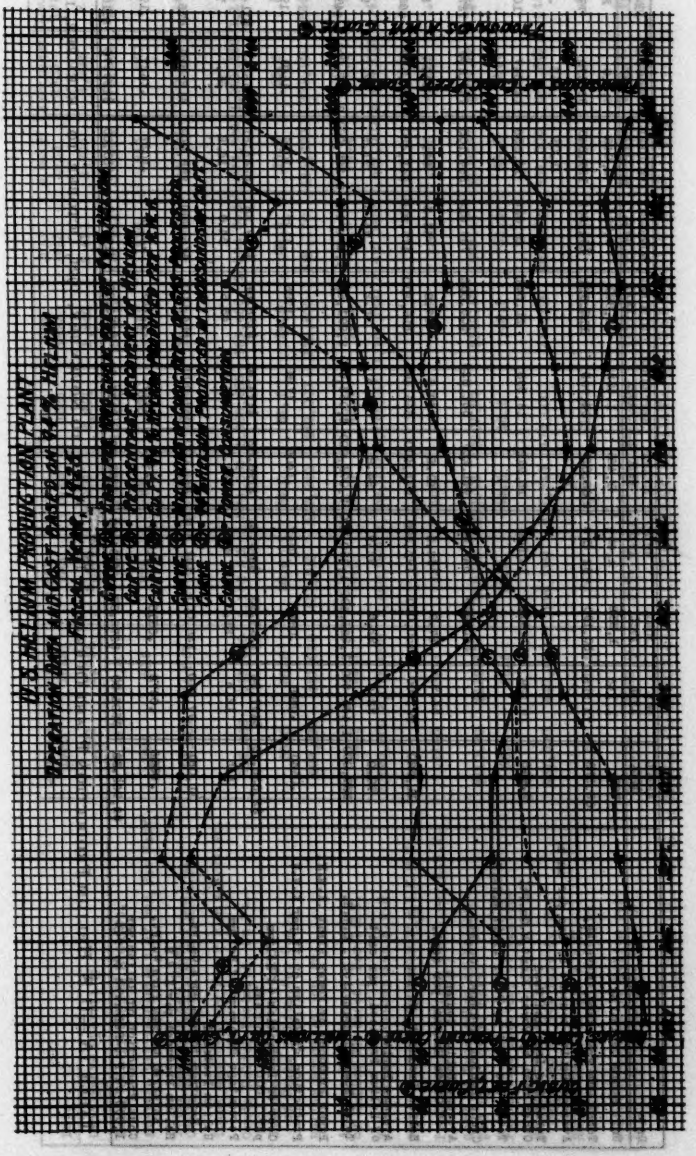


FIGURE 1.

Data	APRIL 1921	MAY 1921	JUNE 1921	JULY 1921	AUGUST 1921	SEPTEMBER 1921	OCTOBER 1921	NOVEMBER 1921	OCTOBER 1922	NOVEMBER 1922	DECEMBER 1922
Unit Hours Operating	720	1,148	1,332	1,464	1,465	1,452	2,880	3,096	2,046	2,116	2,132
Total Gas Processed, M.Cu.Ft.	66,378	73,225	80,190	82,214	68,837	64,905	113,032	151,563	127,040	82,379	79,588
Average Helium content of Gas, per cent.									1.05	.853	.794
Total Gas Lost and Used, Other than by Power Plant, M.Cu.Ft.	9,448	19,359	11,915	14,534	15,912	16,598	21,541	29,026	16,256	14,547	10,490
Total Gas Used by Power Plant, M.Cu.Ft.											
Crude Helium Produced, Cu.Ft.			325,625	349,250	441,220	361,525	908,362	769,010	664,105	746,250	727,950
Average Helium Content of Crude Helium, per cent.		54.9	57.9	58.2	60.5	59.5	57.2	58.5	56.	54.	54.
Helium Produced, Cu.Ft.	11,580	99,000	150,000	193,800	291,400	366,400	816,200	473,200	241,420	420,360	482,980
Average Purity of Helium Produced, per cent.	90.0	92.6	92.8	92.9	94.2	94.5	95.2	94.5	91.5	92.1	92.4
Total Power Consumption . K.W.H.	539,000	1,439,200	1,826,400	1,850,400	2,075,000	2,065,000	2,991,600	4,214,400	2,262,000	2,536,000	2,448,800
Power Produced by Power Plant .K.W.H.											
Production of Helium, cu.ft. per K.W.H.	.0540	.0688	.0822	.1047	.1405	.1788	.1725	.1122	.1066	.1660	.1972
Cost of Production of Helium per 1,000 cubic feet	$2,856.29	$497.54	$257.95	$250.32	$198.83	$174.40	$136.69	$169.35	$243.61	$156.61	$114.29
Percentage Recovery of Helium	1.5	12.5	23.0	29.0	40.0	55.5	45.5	54.0	14.8	55.	70.
Helium Produced - Referred to 98% Pure, Cu.Ft.	11,080	97,500	148,000	191,200	292,000	369,000	524,000	476,000	235,000	412,500	475,000
Helium Produced per K.W.H. Referred to 98% Pure, Cu.Ft.	.0086	.0678	.0810	.1033	.1410	.1788	.1753	.1129	.1037	.1627	.1938
Cost Production of Helium per 1,000 Cu.Ft.-Referred to 98%	$2,968.55	$508.00	$572.00	$384.00	$198.80	$173.50	$156.00	$166.80	$250.00	$159.75	$116.25

TABLE 1A.—U. S. HELIUM PRODUCTION PLANT OPERATING DATA, APRIL TO NOVEMBER, 1921, OCTOBER TO DECEMBER, 1922.

Date	JANUARY 1923	FEBRUARY 1923	MARCH 1923	APRIL 1923	MAY 1923	JUNE 1923	JULY 1923	AUGUST 1923	SEPTEMBER 1923	OCTOBER 1923
Unit Hours Operating	1,191	2,036	2,469	2,969	2,251	1,840	2,621	2,705	2,555	2,575
Total Gas Processed, M.Cu.Ft.	78,678	77,507	97,661	115,439	101,267	84,527	112,221	115,301	98,984	104,350
Average Helium Content of Gas, Per Cent.	.864	.815	.856	.808	.959	.852	.938	.915	.950	.950
Total Gas Lost and Used Other than by Power Plant, M.Cu.Ft.	14,513	14,567	17,341	13,972	12,885	12,724	12,167	12,709	9,958	16,065
Total Gas Used by Power Plant, M.Cu.Ft.										
Crude Helium Produced, Cu.Ft.	665,925	575,250	1,173,360	1,334,080	1,008,655	751,400	1,086,845	1,074,688	1,028,550	1,169,700
Average Helium Content of Crude Helium, Per Cent.	55.0	55.8	55.4	55.0	54.5	54.6	55.2	56.0	55.6	57.2
Helium Produced, Cu.Ft.	458,280	440,090	604,080	627,940	472,000	522,800	568,170	485,820	521,140	656,140
Average Purity of Helium Produced, Per Cent.	93.7	94.1	94.1	94.0	93.5	94.1	94.0	94.4	94.2	95.6
Total Power Consumption K.W.H.	2,180,000	2,072,000	2,460,000	2,854,000	2,171,000	1,766,000	2,427,000	2,629,000	2,424,000	2,415,000
Power Produced by Power Plant, K.W.H.										
Production of Helium, cu.ft. per K.W.H.	.2160	.2124	.2455	.2216	.2174	.1628	.2382	.1847	.1525	.2640
Cost of Production of Helium per 1,000 cubic feet	$110.78	$122.17	$90.66	$95.38	$109.60	$148.50	$89.47	$109.67	$188.68	$82.08
Percentage Recovery of Helium	65.4	65.8	69.6	64.5	45.5	40.9	52.5	45.5	51.7	64.0
Helium Produced - Referred to 94% Pure Cu.Ft.	457,000	440,600	604,700	627,940	470,000	325,200	568,170	488,000	521,800	648,000
Helium Produced per K.W.H. Referred to 94% Pure Cu.Ft.	.2156	.2125	.2457	.2216	.2162	.1830	.2382	.1856	.1526	.2687
Cost Production of Helium per 1,000 Cu.Ft. Referred to 94%	$111.00	$122.00	$90.80	$95.38	$110.10	$148.20	$89.47	$109.15	$186.80	$80.80

TABLE 1B.—U. S. HELIUM PRODUCTION PLANT OPERATING DATA, JANUARY TO OCTOBER, 1923.

Date	NOVEMBER 1923	DECEMBER 1923	JANUARY 1924	FEBRUARY 1924	MARCH 1924	APRIL 1924	MAY 1924	JUNE 1924	JULY 1924	AUGUST 1924
Unit Hours Operating	2,520	2,619	3,627	2,868	2,599	2,978	2,696	3,100	2,506	2,226
Total Gas Processed, M.Cu.Ft.	108,188	107,778	151,741	140,855	119,188	124,857	128,065	147,458	128,160	125,697
Average Helium Content of Gas, Per Cent.	.860	.952	1.055	1.054	1.058	1.045	1.040	1.045	1.004	1.016
Total Gas Lost and Used Other than by Power Plant, M.Cu.Ft.	17,377	23,686	38,995	37,405	27,514	23,847	16,857	15,074	15,538	13,958
Total Gas Used by Power Plant, M.Cu.Ft.					1,890	4,174	12,950	12,446	8,904	11,561
Crude Helium Produced, Cu.Ft.	1,192,335	1,203,475	1,654,900	1,547,595	1,289,500	1,364,565	1,354,645	1,589,140	1,345,100	1,202,155
Average Helium Content of Crude Helium, Per Cent.	56.5	55.5	55.1	55.7	56.5	54.5	55.6	56.0	54.4	55.6
Helium Produced, Cu.Ft.	690,065	731,690	1,014,740	843,180	712,270	725,110	756,640	830,060	696,300	667,060
Average Purity of Helium Produced, Per Cent.	94.7	94.1	94.0	94.5	94.4	93.7	93.8	93.5	93.9	94.5
Total Power Consumption, K.W.H.	2,186,000	2,529,000	3,396,000	2,852,000	2,286,000	2,186,000	2,551,500	3,010,000	2,563,500	2,374,000
Power Produced by Power Plant, K.W.H.					106,000	278,250	682,250	646,000	445,500	705,500
Production of Helium, cu.ft. per K.W.H.	.2700	.2890	.2990	.2955	.3110	.3315	.3030	.2760	.2278	.2470
Cost of Production of Helium Per 1,000 cubic feet	$85.14	$73.65	$65.85	$69.05	$70.74	$65.55	$67.15	$61.35	$83.97	$77.22
Percentage Recovery of Helium	63.6	68.6	59.6	54.8	54.5	52.2	54.0	50.3	41.0	45.5
Helium Produced - Referred to 94% Pure Cu.Ft.	695,000	735,000	1,014,710	848,000	715,000	724,000	765,000	825,000	695,700	690,000
Helium Produced per K.W.H. Referred to 94% Pure Cu.Ft.	.2720	.2900	.2990	.2985	.3125	.3310	.3023	.2740	.2274	.2488
Cost Production of Helium per 1,000 Cu.Ft. Referred to 94%	$84.50	$73.55	$65.85	$68.70	$70.50	$65.70	$67.20	$61.80	$84.10	$76.81

TABLE IC.—U. S. HELIUM PRODUCTION PLANT OPERATING DATA, NOVEMBER, 1923, to AUGUST, 1924.

Date	SEPTEMBER 1924	OCTOBER 1924	NOVEMBER 1924	DECEMBER 1924	JANUARY 1925	FEBRUARY 1925	MARCH 1925	APRIL 1925	MAY 1925	JUNE 1925
Unit Hours Operating	2,630	2,370	2,148	1,412	717	702	745	1,041	745	1,279
Total Gas Processed, M.Cu.Ft.	145,790	139,698	138,837	112,080	97,506	93,145	97,007	127,456	115,991	149,246
Average Helium Content of Gas, Per Cent.	1.010	.977	.998	.979	1.003	1.000	1.005	1.024	1.016	1.009
Total Gas Loss and Used Other than by Power Plant, M.Cu.Ft.	11,530	11,088	13,724	14,035	14,293	13,808	9,741	10,535	8,190	7,564
Total Gas Used by Power Plant, M.Cu.Ft.	9,091	13,176	12,415	9,000	19,861	18,533	20,047	22,799	20,605	27,392
Crude Helium Produced, Cu.Ft.	1,563,960	1,472,325	1,516,100	1,451,120	1,767,710	2,061,462	2,680,789	3,258,670	2,811,500	4,403,780
Average Helium Content of Crude Helium, Per Cent.	55.7	55.1	59.2	45.0	36.0	34.0	30.0	30.2	32.0	28.0
Helium Produced, Cu.Ft.	615,929	795,830	806,300	593,100	658,760	712,860	809,650	993,920	902,800	1,228,500
Average Purity of Helium Produced, Per Cent.	94.5	94.2	94.5	95.8	95.2	95.0	94.0	95.6	95.8	95.6
Total Power Consumption, K.W.H.	2,766,750	2,587,650	1,898,650	1,250,230	917,000	809,350	878,000	1,017,300	923,500	1,241,000
Power Produced by Power Plant, K.W.H.	485,500	744,500	706,250	404,750	808,750	819,500	881,000	1,008,500	923,500	1,241,500
Production of Helium, Cu.Ft. per K.W.H.	.2950	.3072	.4245	.4740	.718	.882	.922	.977	.977	.990
Cost of Production of Helium Per 1,000 cubic feet	$62.98	$62.03	$55.48	$70.15	$52.09	$36.26	$31.61	$28.20	$31.75	$24.30
Percentage Recovery of Helium	52.4	54.8	55.0	51.8	64.2	72.8	78.2	71.5	73.1	72.3
Helium Produced - Referred to 94% Pure, Cu.Ft.	821,500	797,500	810,000	605,000	667,300	720,000	809,650	988,400	900,880	1,225,270
Helium Produced per K.W.H. Referred to 94% Pure, Cu.Ft.	.2970	.3080	.4255	.4840	.728	.890	.922	.972	.975	.986
Cost Production of Helium per 1,000 Cu.Ft. Referred to 94%	$62.65	$61.90	$55.18	$68.85	$51.43	$35.86	$31.61	$28.32	$31.80	$24.40

TABLE 1D.—U. S. HELIUM PRODUCTION PLANT OPERATING DATA, SEPTEMBER, 1924, TO JUNE, 1925.

cost was made in spite of the handicap that only sufficient gas was available to operate the Plant at one-third capacity. Operation continued steadily throughout April with a little extra gas available, which permitted the operation of a second production unit for about ten days. This increased production sufficiently to cut the cost down to $28.00 a thousand cubic feet. The Plant operated at one-third capacity in May, but in June there was sufficient gas to operate at two-thirds capacity for most of the month, and the cost was reduced to $24.00 per thousand, which is the lowest price at which Helium has ever been produced.

Figure 1 shows graphically the main operation data and cost of the production of Helium, reduced to a standard 94 per cent pure, for the fiscal year 1925. Tables 1A, B, C, and D give the same data in tabular form and in more detail for the complete operating period from April, 1921, to June, 1925, inclusive. Table 2 covers the operating data of the Power Plant

Date - - 1925	February	March	April	May	June
Number of Units Operating	3-4-5-6	3-4-5	3-4-5-6	3-4-5-6	3-4-5-6
Unit-hours Operating	2645	2785	3062	2815	4000
Maximum Demand KWH	1775	1500	2000	2100	2150
Minimum Demand KWH	300	740	880	500	925
Average Demand KWH	1220	1184	1400	1241	1739
Total Output KWH	819500	881000	1008500	923500	1241500
Rated Capacity per Unit KWH	512.5	512.5	512.5	512.5	512.5
Output per Unit-hour KWH	309.8	316.3	309.8	328.0	310.5
Unit-hour Load Factor, Percent.	99.14	101.22	99.14	104.96	99.36
Plant Load Factor, Percent.	55.75	54.13	64.03	56.75	79.45
Total Gas Consumed, M Cu.ft.	18553	20047	22800	20605	27592
BTU per Cubic Foot of Gas	700	700	700	700	700
Consumption BTU per KWH Output	15830	15928	15825	15619	15847
Overall Efficiency, without Economizers	21.67	21.44	21.58	21.85	21.62
Overall Efficiency, including Economizers	41.78	41.53	41.80	42.34	41.70

TABLE 2.—U. S. HELIUM PRODUCTION PLANT, POWER PLANT OPERATING DATA.

for the five months that the full plant was in operation and furnishing all of the power required by the Helium Plant. Table 3 gives an analysis of the costs of power production and the actual saving over purchased power for the same period.

PARTICULARS	FEBRUARY		MARCH		APRIL		MAY		JUNE	
	Total Cost	Cost Per KWH	Total Cost	Cost Per KWH	Total Cost	Cost Per KWH	Total Cost	Cost Per KWH	Total Cost	Cost Per KWH
OPERATION:										
Fuel	$3,428.61	$0.004184	$3,708.64	$0.004210	$4,217.80	$0.004183	$3,811.98	$0.004126	$5,067.50	$0.004082
Labor	990.94	.001208	1,107.80	.001257	1,031.28	.001025	1,090.62	.001179	1,044.82	.000840
Oil	411.48	.000502	528.02	.000600	522.25	.000519	587.40	.000636	941.84	.000758
MAINTENANCE										
Material	$ 139.13	$0.000171	$ 144.38	$0.000164	$ 132.94	$0.000131	$ 133.82	$0.000144	$ 381.97	$0.000307
Labor	449.48	.000548	360.60	.000409	313.04	.000310	405.52	.000444	754.96	.000608
GROSS COST	$5,419.64	$0.006613	$5,849.44	$0.006640	$6,217.31	$0.006168	$6,029.34	$0.006529	$8,191.09	$0.006595
Credit for Steam furnished	$1,205.00	$0.001470	$1,205.00	$0.001368	$1,205.00	$0.001195	$1,205.00	$0.001305	$1,205.00	$0.000970
NET COST	$4,214.64	$0.005143	$4,644.44	$0.005272	$5,012.31	$0.004973	$4,824.34	$0.005224	$6,986.09	$0.005625
COST TO PURCHASE POWER	$12,424.80		$12,822.10		$13,748.00		$12,948.30		$15,332.94	
COST TO PRODUCE (Net Cost, above)	4,214.64		4,644.44		5,012.31		4,824.34		6,986.09	
NET SAVING	$ 8,210.16		$ 8,177.66		$ 8,735.69		$ 8,123.96		$ 8,346.85	
NET SAVING PER 1,000 Cu.Ft. Helium	$ 11.52		$ 10.10		$ 8.79		$ 9.00		$ 6.79	

TABLE 3.—U. S. HELIUM PRODUCTION PLANT, OPERATING DATA OF POWER PLANT FROM FEBRUARY 1, 1925, TO JUNE 30, 1925.

The gas processed at the Helium Plant at Fort Worth comes from the Petrolia Field in Clay County, Texas, and is brought in through the Government owned ten-inch gas pipe line, ninety-five miles long. At the time this line was laid, and the Helium Plant established, the Petrolia Field was the most potential one found, but the field has been producing for several years, and at this time is nearly exhausted. The Bureau of Aeronautics, which took over the Helium activities of the Navy when the Bureau was established in September, 1921, and the members of the Army and Navy Helium Board have advocated bringing in the gas from the virgin field at Nocona, Texas, since the fall of 1923, at which time it was appreciated that the Petrolia Field was well on the decline. A matter of thirty-one (31) miles of pipe line is all that is necessary to connect the Nocona Field to the present pipe line in the vicinity of Bowie, Texas. This gas is as high in Helium content as the Petrolia gas and due to its composition will be a gas much easier and much cheaper to process. The Navy is suffering from the lack of sufficient Helium to carry on its airship program, and this shortage is due only to the fact that the necessary funds were not available. In August, 1924, the Army and Navy Helium Board finally approved the Nocona Pipe Line Project and steps were taken to accomplish this long needed project when it was discovered that Congress had not made a specific appropriation for such a contingency, and the pipe line foundered on a legal technicality. During the second session of the sixty-eighth Congress the necessary authority for the line was obtained for the Navy Department, and had not the Bill transferring the Helium activities to the Bureau of Mines been passed by the same Congress the installation of this line would probably have commenced in July, 1925. If the Nocona Field had been reserved for the purpose of extracting Helium, when first advocated by the Bureau of Aeronautics, the output of the Helium Plant for June, 1925, would have been two million cubic feet instead of one and a

quarter, and the cost would have been about $15.00 a thousand cubic feet instead of $24.00.

Helium has a distinct advantage over Hydrogen for use in airships, in that after it has been contaminated with air in service it can be repurified, whereas Hydrogen under the same circumstances has to be thrown away. For this purpose there was constructed at the Naval Air Station, Lakehurst, New Jersey, a Helium Purification Plant, the only one in operation in the world. This process costs about two dollars ($2.00) a thousand cubic feet, and recovers about eighty per cent (80 per cent) of the Helium placed in the ship and not lost during flight, and thereby reduces the ultimate cost of Helium to a value approximately commensurate with the cost of Hydrogen.

In view of the original higher cost of Helium and the possibility of reclaiming it after it has been in use, every precaution is taken to reduce the losses during the flight of an airship. To accomplish this the Bureau of Aeronautics constructed what is known as exhaust water recovery units and installed them on the U. S. S. *Shenandoah* and *Los Angeles*. This device recovers water from the engine exhaust which is retained on board the ship to compensate for the weight of fuel consumed. Every pound of gasoline consumed in the engines will produce about one and a quarter pounds of water vapor in the exhaust, so in order to completely compensate for the total weight of fuel consumed the exhaust condensers have to be only about eighty per cent (80 per cent) efficient. By thus maintaining a constant loading of the ship, the necessity for valving Helium during flight is reduced to a minimum. The data and experience gained from the operation of the first type of units on the two airships will soon lead to a more efficient design which will function in all kinds of weather and produce the required amount of water for ballast.

Never before has any industry been called upon to transport such vast quantities of gas such long distances. With the

advent of Helium, and the Purification Plant at Lakehurst, there developed the need of a large gas-tight storage, and from a military point of view, preferably one underground where its location could not be seen from the air. In 1923 there was purchased from the Bethlehem Steel Company, for the Naval Air Station at Lakehurst, New Jersey, the first large high pressure gas storage in existence in this country. It consists of ten steel cylinders about forty-six (46) feet long and four (4) feet in diameter made by hollow forging from a solid billet and the ends closed with blank flanges secured by studs screwed into the ends of the cylinders. This battery of cylinders is placed underground and concrete head chambers are provided at both ends for the purpose of inspection and of charging and discharging. This storage is designed for a pressure of 2500 pounds per square inch, and under this pressure each cylinder is capable of holding 65,000 cubic feet of Helium measured at atmospheric pressure and a temperature of 70 degrees F. From this start, the engineers of the Bureau of Aeronautics, working with those of the Bethlehem Steel Company, evolved the Helium Tank Car—three massive tanks mounted on one railroad car, capable of carrying 200,000 cubic feet of Helium under a pressure of 2000 pounds per square inch.

At the time the Helium industry came into existence the only method of transporting gas was in small high pressure cylinders as used for oxygen and hydrogen. Each of these cylinders will hold 178 cubic feet of Helium under a pressure of 1800 pounds per square inch. Each cylinder weighs about 130 pounds and holds about two (2) pounds of Helium. While this form of container is very satisfactory for shipping small quantities of gas, it is unsatisfactory and very costly for large quantities at long distances. It took 13,000 of these small cylinders to fill the U. S. S. *Shenandoah* at her first inflation. In spite of the great care which is taken to insure the tightness of the valves on these cylinders, the leakage from

the time they are filled until they are emptied at Lakehurst, is from ten to twenty per cent. It takes a crew of men about two days to unload a car of about 600 of these cylinders and prepare them for discharge after the carload is received at Lakehurst. Whereas with the Tank Car, one man in five minutes after the car arrives can be ready to supply twice as much Helium as contained in the 600 small cylinders. It was the realization of the inadequacy of these cylinders and the potential possibilities of the large storage cylinders, that led to the design of the Tank Car. This car consists of three large steel tanks, forty (40) feet long and four (4) feet nine (9) inches in diameter, mounted on a special railway truck. The cylinders are made of a special alloy steel, hollow forged from a solid billet to walls four (4) inches thick. After forging, the tube is machined inside and out to reduce the wall thickness to about two (2) inches, in order to reduce the weight of the total car to about 200,000 pounds. The cost of transportation will be cut down materially; a big saving in the cost of handling will be effected, and the loss in shipment of about ten to twenty per cent of the Helium will be eliminated as soon as an appropriation can be obtained to purchase enough of these cars to meet the requirements of the service. Due to the recent improvements in the production of Helium, it now costs more to handle and ship the gas than it does to produce it.

The Army and Navy have used the twenty-four (24) million cubic feet of Helium, which has been produced at the Helium Plant since it commenced operation in April, 1921, until July, 1925, in lighter-than-air ships, and the Navy could have used more if it had been available. As this activity increases, the demand for Helium will increase, reaching about twenty million cubic feet a year for both services in the near future. The inauguration of commercial airship transportation will double this demand, and after its practicability is definitely proven, this field will probably call for at least fifty

(50) million cubic feet a year. It is very much to the interest of the Navy to encourage this new enterprise, as the commercial air fleet of the future will be a very important adjunct in case of war, especially so as there will be ample trained pilots available and the ships will be filled with Helium. It is the production of the Helium for complete inflation that takes a long time and large production plants; the make-up Helium required to take care of operation losses is relatively small and a large fleet of airships can be maintained by the production of relatively small plants.

The Navy Department and the Bureau of Mines are now actively engaged in an extensive program of experiments with men using an artificial air composed of various mixtures of Helium and Oxygen as a substitute for normal air for use in diving, mining and tunnelling work. The preliminary results are very encouraging. Helium is without odor or taste, and has physical properties which promise to be of interest physiologically. The substitution of Helium for the nitrogen ordinarily present in the air we breathe has been found to result in an atmosphere which is as respirable as that provided by nature, and at the same time will reduce the hazard of caisson disease. The results obtained indicate that Helium not only has the advantage of being less soluble than nitrogen, but also has the advantage of diffusing more rapidly in the blood and other body fluids and tissues which results in rapid elimination of the gas from the tissues during decompression. So from a product developed as a war-time necessity there has been created a need for humanitarian purposes.

The writer has recommended to the Bureau of Engineering that experiments be conducted with Helium filled searchlights on account of the following properties of that gas: high thermal conductivity; low dielectro cohesion; inert, and will produce a yellowish light. Some experimental work has been done in the use of Helium in a closed system for cooling large electrical generators and motors, and as soon as Helium is

more plentiful its usefulness will increase rapidly. This should permit a large reduction in the amount of copper in the machines and increasing the load on the machines already built with a consequent increase in efficiency. Hydrogen is a little better suited for this purpose, but the danger of explosion is too great to permit its use.

The future supply of Helium for the various demands which will be made for it lies primarily with the great Petroleum Industry of this country, against which falls the necessity of discovering new Helium bearing natural gas fields, and conserving this gas in the ground until it is required to furnish the Helium which is contained therein. It is an economic impossibility to extract Helium and store it in containers. The only practical method of conserving this important gas is to leave it in the ground in its natural state until it is required for immediate use. The responsibility of the Petroleum Industry in this respect was laid before it at the general session of the American Petroleum Institute at Fort Worth, Texas, in December, 1924, by the Navy Department, and was given much consideration by that Institute.

At the present time there are six well defined areas at some distance apart which are developed as producing or potential natural gas fields, and sufficiently rich in Helium to warrant the commercial extraction thereof. The Government Helium Plant at Fort Worth is tapping one of these areas, and a second, located on Federal land, has been set aside as a reserve for National emergency. A third area is also on Federal land, but has not been withdrawn. That leaves three areas rich in Helium and not on Federal lands, which are already known, and are producing gas which is available for the extraction of Helium by commercial concerns. The Bureau of Aeronautics, through its Chief of Bureau, Rear Admiral W. A. Moffett, has already taken the stand that the commercial production of Helium is a necessity, and it will be encouraged by that Bureau. The Bill passed by the last session of Con-

gress, which transferred the Helium activities from the Army and Navy Helium Board and the Navy Department, also specifically requires the Government to purchase Helium from private concerns as soon as they are in a position to furnish it at a cost comparable with what it costs the Government to produce it at the Fort Worth Plant. The future supply of Helium to the various consumers is, therefore, placed directly up to the great industrial organizations of this country.

STARTERS AND CONTROLLERS FOR NAVY D. C. MOTORS—TYPES AND PRINCIPLES OF OPERATION.

By C. HUEY, CIVIL MEMBER.

All direct current motors in the Navy over one horsepower are required to be started by a current limiting device. This device is known as a "starter" or "controller." These terms, although descriptive in themselves, are often used synonomously and need therefore be defined.

STARTERS AND CONTROLLERS DEFINED.

A "starter" is a device for limiting the current until the speed of the motor has increased to such a value that the motor can safely be connected directly to the line voltage. The primary function of the "starter" is to get the motor "on the line." The addition of protective or safety devices such as an overload release, a no-voltage release, fuses, line switches, etc., does not change the correct application of the term "starter."

A "controller" is a device for starting, stopping and reversing, or controlling the speed of a motor. The primary function of the "controller" is to permit the change in direction of rotation of the motor, or to regulate or vary the speed of the motor other than that caused by the external load or electrical characteristics of the motor. The controller includes the functions of a starter, and the addition of protective devices does not change the application of the term "controller."

GENERAL CLASSES OF STARTERS AND CONTROLLERS.

Starters and controllers may be divided as to degree of manual operation, into three general classes:

(*a*) Non-automatic or hand operated.
(*b*) Semi-automatic.
(*c*) Full automatic.

They may be further divided as to type or form into the following classes:

(*a*) Panel type.
(*b*) Drum type.
(*c*) Contactor type.

Again they may be further classified as to form of enclosure as follows:

(*a*) Open.
(*b*) Non-watertight.
(*c*) Watertight.
(*d*) Dust-tight.
(*e*) Gas-tight.
(*f*) Semi-protected.
(*g*) Drip-proof.
(*h*) Protected.
(*i*) Water proof.

The semi-automatic and full automatic starters and controllers may be divided as to method of acceleration, into the following classes:

(*a*) Counter electro motive force.
(*b*) Current limit.
(*c*) Time limit.

The success or usefulness of motors in Naval service depends to a considerable extent on the proper selection of the class of starter or controller best suited for the particular purpose. The scope of this article will cover the advantages and disadvantages

of the various types or classes, and will give the diagrammatic wiring schemes, arrangement of protective devices, and a general discussion of the subject as an aid to the operator or designer of the starters or controllers for Navy motors. As starters and controllers have duty to perform in common, *i.e.*, that of accelerating the motor, from rest, starters only will be discussed, in this article, as the general operating principles will be found to apply equally to controllers.

NON-AUTOMATIC OR HAND STARTER.

Panel Type. The meaning of "panel" type or sometimes called face plate type, as at present applied to Navy starters, is the result of evolution and specifications. The actual form as now required for a non-automatic panel type starter, embodies an insulating panel on the front of which is mounted contact segments, a starting arm, a line switch, enclosed fuses, an overload device, a no-voltage release, and necessary terminals for incoming and outgoing wires. The resistors are invariably mounted on the back of the panel for the sake of compactness. This type is probably the simplest form of starter, and for this reason and due to its low cost, it is used on a large number of installations for motors not less than one horsepower nor more than 12 horsepower, where the motors are infrequently started, and where no speed control is required, such as for ice-cream freezers, dough mixers, dish washing machines, potato peelers, motor generators for interior communication sets, laundry machinery, and machine shop tools. The limiting factor in the use of this form of starter, is in the ability of the starting contacts to satisfactorily withstand the service of the particular application.

In order to obtain a clearer insight into the features incorporated in the panel non-automatic starter, Figure No. 1 is illustrative of the usual electrical circuits to be expected.

Explanation. Panels of the above type usually have five terminals; viz., two for line wires (L 1 and L 2), one for

armature and shunt field in common (A + F), one for the opposite terminal of the shunt field (F), and one for the opposite terminal of the armature (A). The motor will run in its predetermined direction of rotation due to the internal connection of the motor. The armature circuit of the starter and motor are shown in heavy lines. This circuit is simple, but contains important features; such as, contact segments or buttons renewable from the front of the panel, and provisions

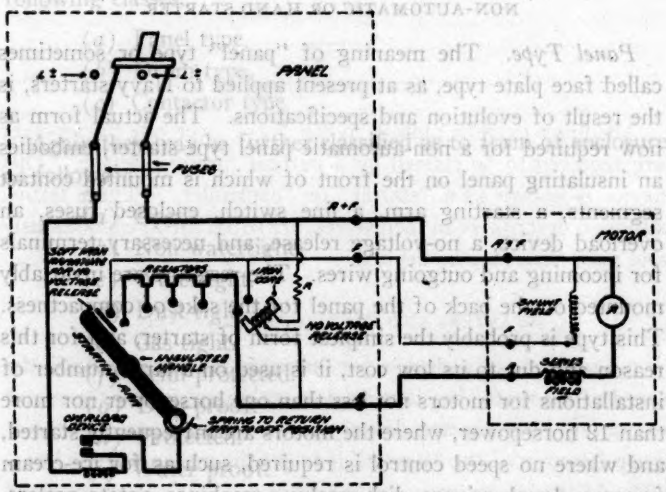


FIGURE 1.
HAND STARTER, PANEL TYPE.

for preventing objectionable arcing at the first starting contact. This latter feature is preferably done by means of a series magnetic blow-out coil mounted behind the first starting contact. A quick release break, by means of a spring and tumbler arm, is sometime provided for the purpose, but on account of the additional moving parts, it is not preferred. The shunt field is required to receive full line volts or excitation on starting and

in the running position of the starting arm. This feature may be accomplished by a field contact button at the first starting point, and by a second field contact button at the running position in parallel with the first button. On the intermediate positions of the starting arm, the shunt field feeds from the starting arm back through the starting resistors to the first contact button. Due to the fact that the resistors are very low in ohmic resistance, the shunt field receives very nearly full line voltage at all intermediate points of starting. It is desirable that the starting contacts be so arranged that the circuit be made on the armature and shunt field at the same instant. This will give a start at full load current or greater, without shock or jar to machinery, for the reason that, due to the large induction of the shunt field compared to the armature, the shunt field strength builds up slowly and the torque of the armature correspondingly builds up slow. A series or compound motor, started in the same way will build up its torque much faster. No motor, however, will build up its field strength instantly, so that it is not likely that any type of motor will give a hard shock to the machinery if it is started with zero field strength.

An additional feature is always incorporated in the wiring scheme to permit the inductive voltage of the shunt field to discharge into the armature of the motor when the line switch is pulled or if the starting arm is returned to the off position. This feature is accomplished by having the shunt field circuit always in connection with the armature circuit in the off position of the switch or starting arm, as shown in the diagram. If the shunt field circuit were to be connected otherwise, without some form of auxiliary discharge circuit, and inductive voltage would be generated at the instant of field break which might be of amount sufficient to puncture the insulation of the shunt field.

Drum Type. The drum type (or sometimes called cylinder type), is used for motors over 12 horsepower; or for less than 12 horsepower, where the arcing conditions are sufficiently

destructive to cause the sliding contact of a panel type starter to jam or freeze after short service. The drum type consists essentially of a row of contact fingers mounted on a frame work, and electrically insulated from it and from each other. An insulated drum or cylinder, on which are mounted copper segments of different length, is located so that when revolved by means of the starter handle, the segments make contact with proper sequence to complete the electrical circuits to the motor and external starting resistors. The wiring or electrical circuit scheme of drum type starters, as furnished with the apparatus, is usually involved to such an extent that the average operator who can readily trace out and fix in his mind the operating circuits of a panel type starter, looks on the drum type starter as more or less of a mystery box. In order to show clearly the scheme on which the drum type operates, simplified diagrams, Figure 2 and Figure 3, illustrate the relation of the motor, starting resistors and drum type starter in the "off" position, and in the first step of starting.

In Figures 2 and 3, a compound motor is assumed, and for simplicity, only primary devices are shown. The contact fingers are represented by the circles, and the drum segments are rolled out in a plane for diagrammatic convenience. The numbered dotted lines show the different positions of the drum segments which the operator "feels" by virtue of a mechanical star wheel and roller cam as the starter handle is rotated. On the "off" position, with the main switch closed, the motor circuit is open at one end. On the first "running" position, the shunt field is across the line, and the armature and series field are on the line in series with the entire starting resistance. As successive notches of the starter handle advance the segments, the starting resistance is short circuited step by step. When the fifth point is reached, the motor is connected directly across the line.

Contactor Type. The contactor type of starter was developed for heavy duty motors where the panel or face plate type

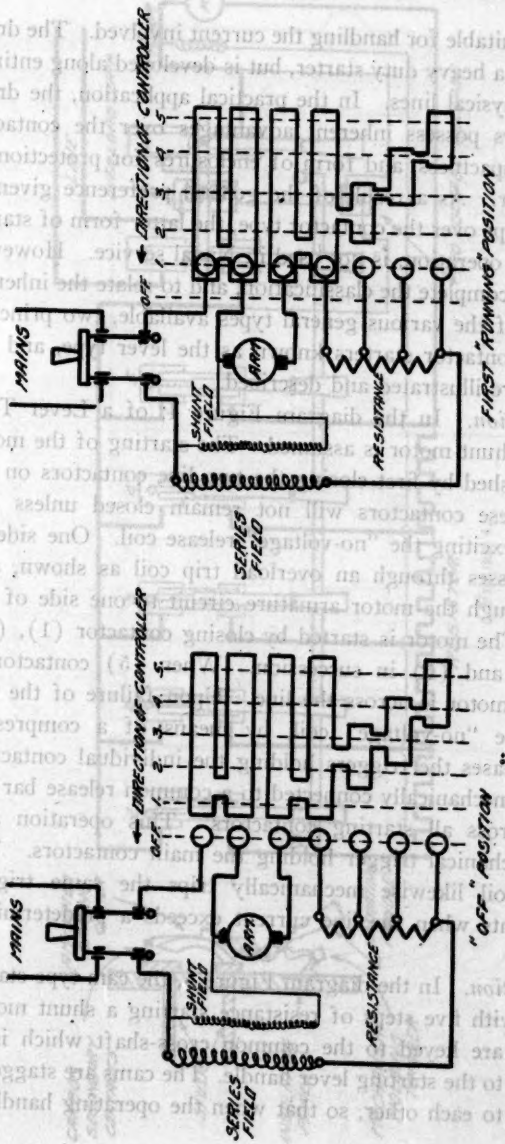


FIGURE 3.

FIGURE 2.

DRUM TYPE, HAND STARTER.

proved unsuitable for handling the current involved. The drum type is also a heavy duty starter, but is developed along entirely different physical lines. In the practical application, the drum type starters possess inherent advantages over the contactor type in compactness, and form of enclosures for protection of the operator. As a result of the general preference given to the drum type over the contactor type, the latter form of starter for manual operation, is not used in Naval service. However, in order to complete the classification, and to relate the inherent principles of the various general types available, two principle forms of contactor starters known as the lever type, and the cam type are illustrated and described.

Explanation. In the diagram Figure 4, of a Lever Type Starter, a shunt motor is assumed. The starting of the motor is accomplished by first closing the two line contactors on the right. These contactors will not remain closed unless line voltage is exciting the "no-voltage" release coil. One side of the line passes through an overload trip coil as shown, and thence through the motor armature circuit to one side of the resistor. The motor is started by closing contactor (1), (2), (3), (4), and (5) in succession. When (5) contactor is closed the motor is across the line. Upon failure of the line voltage, the "no-voltage" coil, by means of a compressed spring, releases the triggers holding the individual contactors which are mechanically connected to a common release bar extending across all starting contactors. This operation also trips a mechanical trigger holding the main contactors. The overload coil likewise mechanically trips the same trigger arrangements when the line current exceeds a predetermined value.

Explanation. In the diagram Figure 5, the cam type starter is shown with five steps of resistance starting a shunt motor. The cams are keyed to the common cross-shaft which is in turn keyed to the starting lever handle. The cams are staggered in relation to each other, so that when the operating handle is

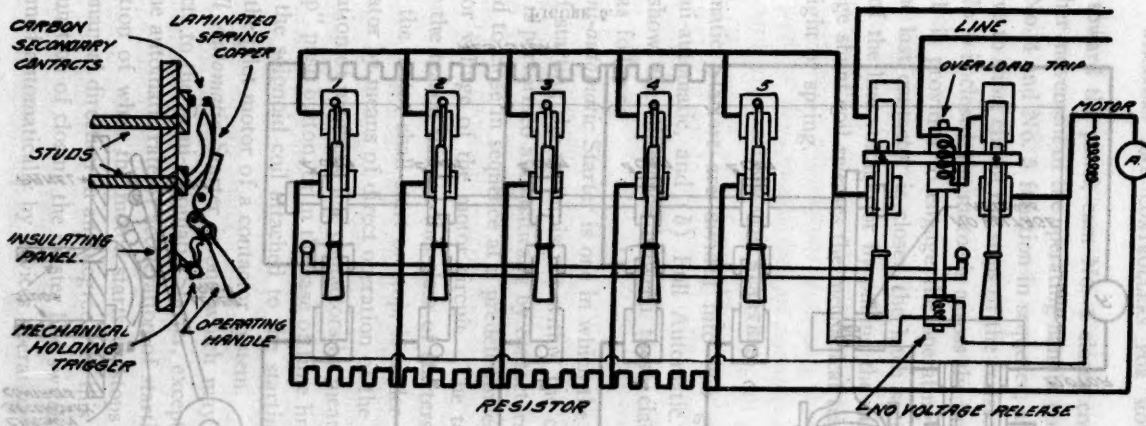


FIGURE 4.

LEVER CONTACTOR TYPE, HAND STARTER.

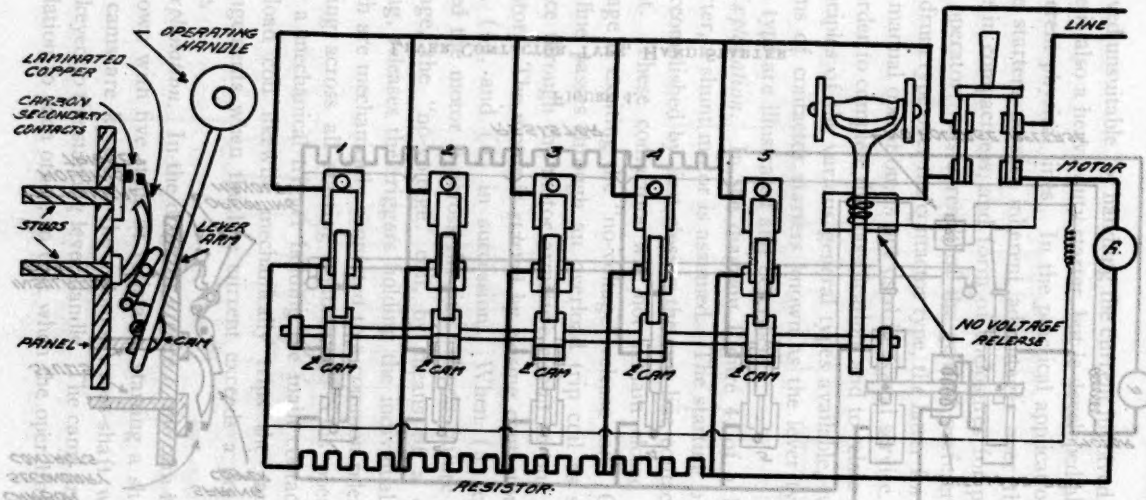


FIGURE 5.

CAM CONTACTOR TYPE, HAND STARTER.

pushed toward the panel, cam No. 1 closes contactor No. 1. By further movement of the operating handle, contactors No. 2, No. 3, No. 4 and No. 5 function in sequence, and when closed they serve to short circuit the steps of the resistor. When contactor No. 5 is closed, the motor is across the line. A mechanical latch is provided to engage the operating lever, so that when the last contactor is closed the latch secures it. Upon failure of the line voltage, or by pulling the main switch, the no-voltage shunt coil releases the mechanical trigger by means of a weight or spring.

AUTOMATIC STARTERS.

Automatic Starters are divided into two general classes, (*a*) Semi-automatic, and (*b*) Full Automatic. In order to clearly show the difference between these classes, they are defined as follows:

A *Semi-automatic* Starter is one in which the rate of acceleration of starting the motor is not within the operator's control, but is performed automatically by contactors designed and calibrated to close in sequence at a predetermined condition of current or voltage of the motor circuit. The term "semi" is used for the reason that on this class of starters, the function of when the motor shall start or stop, is under the control of the operator by means of direct operation of the line switch, or the operation of magnetic line contactors by means of a "start" and "stop" push button, or in the case of time limit devices, by exciting the solonoid coil attached to the starting arm, or by starting the pilot motor of a contactor system.

A *Full Automatic* Starter is one which may be similar in all respects to the Semi-automatic Starter, except that in addition to the automatic rate of acceleration of starting the motor, the function of when the motor starts or stops is performed without manual direction in any degree. In this class of starters the function of closing the master line switch or contactors is performed automatically by devices operating a "start" and

"stop" switch, mechanically connected to high or low level floats in tanks, high or low pressure liquid or gas systems, stop limits for automatic reversing machinery, etc. This operation is also performed electrically by balanced magnetic relays calibrated to a predetermined voltage, such as for use on reserve motor-generators, booster sets, etc.

METHODS OF AUTOMATIC ACCELERATION.

In order to progressively present the Semi-automatic and the Full Automatic classes of starters, the accelerating contactors, which are common to both classes, will be described by first separating the method of acceleration into three groups known as (A) Counter Electro Motive Force, (B) Current Limit, (C) Time Limit. Contactors of the first two groups, such as are used on automatic starters, are operated by the magnetic pull exerted by copper coils wound on a soft iron frame or core; in close proximity to which there is a hinged or pivoted iron armature mechanically connected to levers or toggles which operates to close the contactors when the coils are energized. The method by which the coils are energized are covered by the first two groups mentioned above.

ACCELERATION BY COUNTER ELECTRO MOTIVE FORCE.

The term counter electro motive force is used for the reason that the force employed is the electro motive force, or voltage, generated by the motor by virtue of its armature revolving in its magnetic field. This action is exactly similar to a generator, and the voltage so generated being of the same polarity as the line voltage, it opposes it. Therefore the term "counter" is used in the expression "Counter Electro Motive Force" abbreviated to c. e. m. f. The c. e. m. f. is proportional to the number of turns in the winding of the armature, the field strength and the R.P.M. The first factor is fixed, and the second factor is fixed for shunt motors starting on full field. Therefore the c. e. m. f. generated will vary practically proportionally to the R.P.M. When the motor is at rest, the

c. e. m. f. is naturally zero, and if line volts were to be applied to the armature terminals, the resultant current would be limited only by the very small ohmic resistance of the armature circuit. As the motor starts and accelerates the c. e. m. f. builds up to an amount equal to the applied volts, less the voltage drop due to the ohmic resistance of the armature circuit. In order to determine the current which will flow in the armature circuit under the above conditions it may be said that the motor has two e. m. f.'s, one of which is applied voltage E which is the line voltage if no resistor is used (or the line voltage less the voltage drop across the series resistor when used), and the other, the motor generated e. m. f. which is termed *e*. The current I which will flow at any instant will be

$$I = \frac{E - e}{R + r}$$

where R equals the resistance of the armature circuit and *r* equals the resistance of the resistor circuit.

The application of the principles involved as applied to the c. e. m. f. contactors, is shown by referring to Figure 6 which shows the relation existing between the line volts, terminal volts, and the motor e. m. f. Point 1 is obtained when the main switch or contactor is closed, on either a semi-automatic or full automatic starter, and indicates the motor at rest with current flowing as limited by the resistance of the armature and resistor. When the motor accelerates to point 2, the first step magnetic contactor, which is wound to operate at the voltage of this point, attracts its armature and closes a contactor which short circuits the first step of the resistor as shown in Figure 7. Likewise when the motor further accelerates to point 3, the second step contactor operates from the voltage at this point, and closes a contactor which short circuits the last step of the resistor and applies full line volts to the motor.

It will be noted in Figure 7 showing the scheme of a two step c. e. m. f. automatic starter, that although the main current passes through the contactors, the operating coil itself is

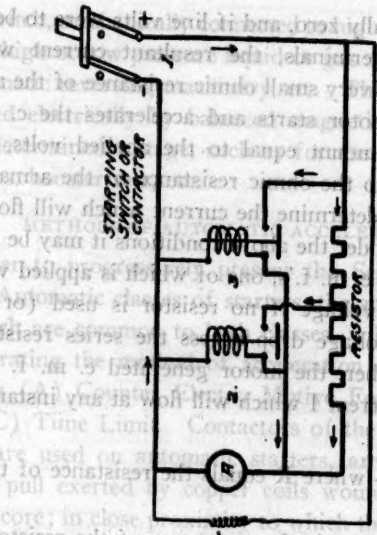


FIGURE 7.

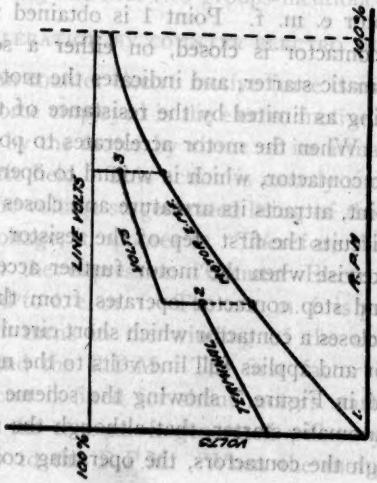


FIGURE 8. COUNTER ELECTRO MOTIVE FORCE, AUTOMATIC, TWO STEP STARTER.

a potential coil, and functions from voltage derived from the motor. For this reason the winding of the coil can be made compactly and at low cost for any size motor. These features are the two chief factors in the use of the c. e. m. f. principle for starters. An analysis of the inherent electrical characteristics at varying line voltage, will be discussed later on, in comparison with the current limit type of starter.

ACCELERATION BY CURRENT LIMIT.

The term "current limit," is used for the reason that the control medium employed is the main current flowing through the armature circuit. This is in contrast to the "counter electro motive force" type, which employs the voltage generated by the motor. The term "limit" is used as descriptive of the fact that the devices employed are designed and adjusted to limit the operation of the accelerating contactors to a current not to exceed a predetermined amount. There are two recognized and accepted current limit types of starters, known as (1) Series (or current) relay; (2) Series (or current) Lockout.

SERIES (OR CURRENT) RELAY METHOD OF ACCELERATION.

This method consists of magnetic contactors with operating coils wound for voltage, in combination with series or current relays operating from the armature circuit current. The voltage wound contactors may be similar in all respects to the c. e. m. f. contactors, except that the coils are wound to operate on line volts instead of motor volts. The relays serve to hold open the voltage supply to the coils of the contactors until the current in the armature circuit is reduced by the acceleration of the motor, to a predetermined amount. A simplified diagram of a series relay type of three step starter is shown in Figure 8.

It will be noted in Figure 8, showing a series relay automatic starter, that the operating coil No. 1 is across the line when the starting switch is closed. The magnetic pull of coil No. 1 attracts and closes the main contactors and the current passes through series relay "B" to the resistor and thence to the motor armature. This operation is the first step of starting and

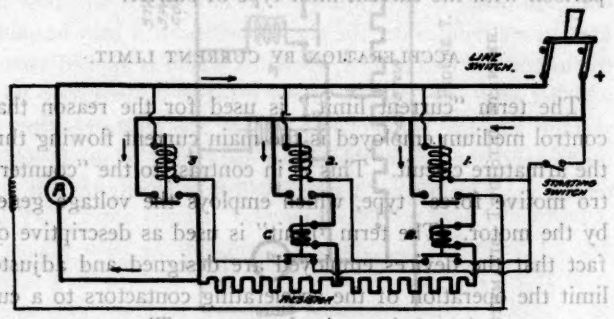


FIGURE 8.
SERIES RELAY, AUTOMATIC, THREE STEP STARTER.

acceleration of the motor. Series relay "B" functions in a peculiar manner as follows:—Before current circulates through it, its armature is held up by a spring so that the auxiliary contacts feeding coil No. 2 are out of circuit. When current reaches series coil "B," the spring is released by a mechanical connection to the plunger of coil No. 1, and the motor current holds up the plunger of the relay "B" until the acceleration of the motor causes a reduction in current to a predetermined amount. At this point the plunger of relay "B" falls, and closes the circuit to the operating coil No. 2 of the second contactor. When contactor No. 2 closes, a portion of the resistor is short circuited, and the second step of acceleration is performed. Relay "C" performs in the same manner as relay "B" and when it falls, contactor No. 3 operates to place the motor across the line.

SERIES (OR CURRENT) LOCKOUT METHOD OF ACCELERATION.

This method consists of magnetic contactors with operating coils wound for armature current. Voltage coils or relays need not be employed in this method. In one method in service, operation is accomplished by predetermined values of armature current in combination with a clever system of utilizing the saturation of the iron in one portion of the magnetic circuit. The operation of one form of this type of contactor can best be understood by reference to Figure 9.

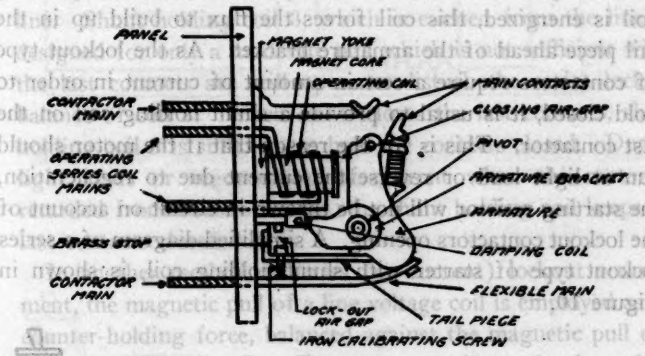


FIGURE 9.

SERIES LOCK-OUT, MAGNETIC CONTACTOR.

Flux or magnetism in the iron is caused by current flowing through the operating coil. This flux passes through the air-gap to the armature of the contactor. Part of this flux passes from the armature through the armature bracket to the magnet yoke, and thence to the magnet core. Another part of the flux passes from the armature through the tail piece to the magnet yoke. The flux through this last circuit exerts a pull which prevents the contactors from closing. The magnetic path through the armature bracket has a small cross-section, so that when the current flowing through the operating coil exceeds a certain value, it becomes saturated and forces the balance of the flux through the tail piece, holding the contactor open. As

the current decrease due to acceleration of the motor, the flux through the tail piece decreases until it is not sufficient to hold the contactor open. The switch can be adjusted to close at a predetermined current value by means of a calibrating screw which changes the hold-out air-gap between the tail piece and the magnet yoke. When the circuit is first completed through the operating coil there is a tendency for the switch to close before the flux in the tail piece is sufficient to lock it open. This tendency is overcome by placing a heavy copper damping coil around a portion of the armature bracket. When the operating coil is energized, this coil forces the flux to build up in the tail piece ahead of the armature bracket. As the lockout type of contactors require a certain amount of current in order to hold closed, it is usual to provide a shunt holding coil on the last contactor. This is for the reason that if the motor should run at light load or reverse the current due to regeneration, the starting resistor will not be thrown in circuit on account of the lockout contactors opening. A simplified diagram of a series lockout type of starter with shunt holding coil, is shown in Figure 10.

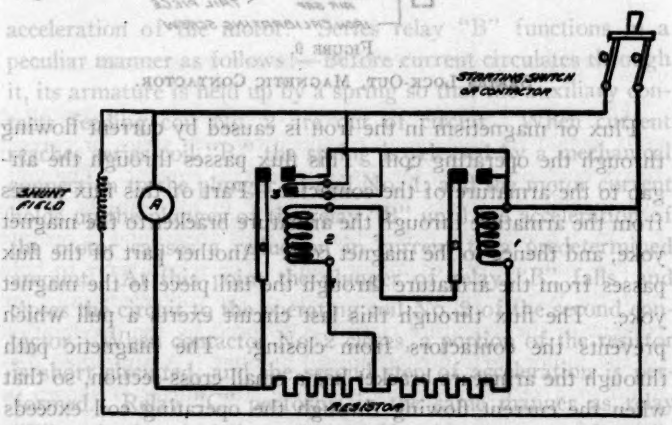


FIGURE 10.

SERIES LOCK-OUT, AUTOMATIC TWO STEP STARTER WITH SHUNT HOLDING COIL.

In the diagram shown in Figure 10, the scheme of operation is as follows: When the main switch is closed, current flows through series coil 1 and through the resistor to the motor armature. The lockout armature of coil 1 holds its contacts open until the acceleration of the motor reduces the current in the line to a predetermined amount. At this point the contacts close and feed the line through series coil 2 and cut out one step of the resistor. When the motor accelerates still further and reduces the current to a predetermined amount, the contact for coil 2 closes and places the motor armature across the line. Shunt holding coil 3, which is excited from the line, is designed for only a moderate pull which is not sufficient when the last contacts are still open, to disturb the series lockout balance of coil 2. When coil 2 closes the last contacts, shunt holding coil 3 is sufficient to hold the contacts closed. During light loads or regeneration of current when the motor is retarded from external causes, the shunt holding coil keeps the last contacts closed.

In another method of series (or current) lockout arrangement, the magnetic pull of a line voltage coil is employed as the counter-holding force, balanced against the magnetic pull of a series current coil. By using a saturated core in the voltage coil, the effect of changing line volts is made negligible, and the general scheme of operation as a series lockout principle is satisfactorily accomplished in a simplified form. The general scheme and relation to the motor circuit as a one step starter is shown in Figure 11.

Still another form of series (or current) lockout arrangement employs a series coil with a saturated core as a counter-holding force, balanced against another series coil with an unsaturated core. At predetermined values of lowering current, the balance of forces acting on the contactor arms is reversed by virtue of the saturated core maintaining its magnetic pull during lowering current, while the unsaturated core effects a reduced magnetic pull proportional to the lowering current. The general scheme of this principle is shown as a one step starter in Figure 12.

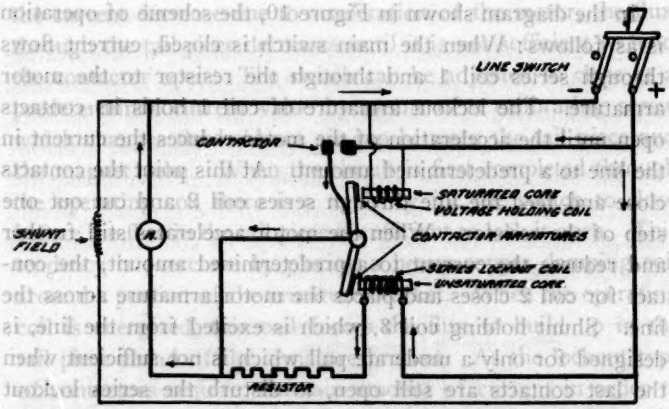


FIGURE 11.

SERIES LOCK-OUT, AUTOMATIC, ONE STEP STARTER, WITH SERIES AND SHUNT COILS.

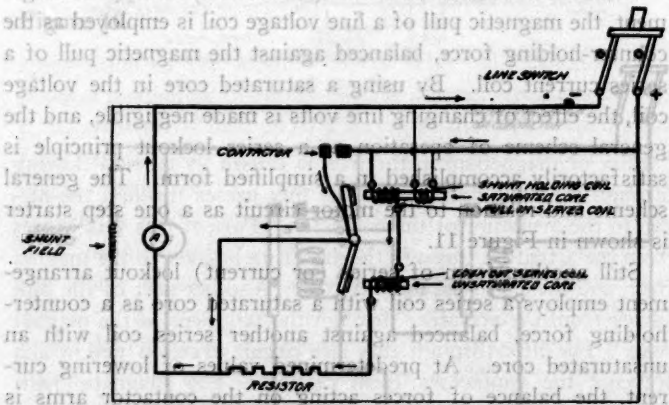


FIGURE 12.

SERIES LOCK-OUT, AUTOMATIC, ONE STEP STARTER, WITH SERIES COILS.

ACCELERATION BY TIME LIMIT.

In this method the short-circuiting of the starting resistance is controlled by means of a solenoid form of shunt coil restrained to a definite time limit by means of a dash pot. This device can be operated directly on the main arm of a face plate starter, or, on contacts which energize individual shunt contactors. Due to the difficulty in maintaining the proper adjustment to overcome friction of moving parts, this form of time limit is not considered satisfactory but has been used in earlier forms of automatic starters. A successful time limit device employs a small pilot motor which is geared to turn a drum controller at a definite speed. The time of acceleration is adjusted by changing the speed of the pilot motor. The drum controller may operate either the main contacts for short circuiting the resistor, or operate auxiliary circuits to energize shunt coil contactors. In either case the acceleration of the motor is smooth under all conditions of load regardless of the amount of current required.

SUMMARY OF TYPES OF AUTOMATIC STARTERS.

The foregoing four schemes of automatic acceleration of motors are the usual ones to be expected both in Naval and commercial service. Opinion differs somewhat as to the best method to use. The practical application for which the controller is designed must be considered in relation to the importance of the motor, the relative cost of the controller, the character of the mechanical load, and the variation of the supply voltage.

The counter e. m. f. method is probably the lowest cost and is suitable for use for motors where the line voltage remains practically normal.

The series relay method is probably the highest cost and for this reason its use is usually considered only for large motors.

It is successfully used for heavy starting duty where the load varies throughout wide limits. This method is not affected by variation in line volts.

The series lockout type is satisfactory for starting duty service where the acceleration is always under load. The last resistance switch is usually provided with a shunt holding coil to keep it from dropping out under light loads. This method is not affected by variation in line volts.

The time element device with Pilot Motor is satisfactory for use where the voltage varies over wide limits, particularly where rapid starting is not necessary. Due to the fact of its inherent slow starting on light loads, and the necessity of a pilot motor, this form of starter is not usual in Naval service, except for special duty such as laundry washing machines, where the time element is successful for driving the machine through a definite cycle of ahead and reverse operation.

SEMI-AUTOMATIC STARTER AUXILIARY DEVICES.

As shown in the preceding paragraphs, the method and types of contactors used for starting duty may be similar for both the semi and the full automatic starters. The additional devices used for the semi-automatic starters consist of such accessories as a two pole line current breaker, a lever main switch, or a start and stop push button or switch. The first two items mentioned above are used by the operator to transfer the line current to the various circuits of the starter, as shown in Figures 7 and 10, and constitutes the first step in the starting operation. On larger motors, where it is not desired that the operator should handle the main line current during the actual starting, an auxiliary push button or switch can be provided as shown in Figure 8, where the operator controls only a small exciting current to the voltage coil of the first contactor. The method is also used for distant control of the starting or stopping of any size of motor. In this arrangement, the main switch is left in the closed position, and the auxiliary start and stop

switch, which is required to carry only a few ampères in current, is installed and operated at a distant point. An additional selective transfer switch is usually mounted on the starter panel to shift the point of starting from local to distant operation. Two schemes of push button start and stop devices are in use, one of which is a two wire and the other a three wire scheme. The first scheme is used when it is desired that the motor automatically re-start upon return of the line voltage after failure. The second scheme is used when it is desired that the motor not re-start unless the operator pushes the start button. The wiring scheme of the above two methods showing the principles involved are shown under Figures 13 and 14.

Explanation. With reference to Figure 13 showing a two wire start and stop push button current, the "start" button is mechanically interlocked with the "stop" button. When the start button is pushed, it completes the voltage circuit to the shunt coil of the magnetic contactor main switch. This circuit remains closed until the stop button is pushed, which operation mechanically opens the start button circuit. In Figure 14, showing a three wire start and stop push button, the start and stop buttons are not mechanically interlocked. The start button is held normally open by means of a spring. When this circuit is momentarily closed, magnetic coil 2 closes the main circuit 1, and in addition, closes auxiliary circuit 3 which parallels the start button through the stop button. The stop button is normally held closed by a spring. Upon release of the start button by the operator, the magnetic coil circuit remains closed through contact 3 and the stop button contacts. Upon failure of the line voltage, or by pushing the stop button, contact 3 is released and the motor will not start again when the line voltage returns, unless the start button is again momentarily closed. If more than one distant control "start" and "stop" button is desired, the "stop" buttons are wired in series, and the "start" buttons are wired in parallel.

FIGURE 14. THREE-WIRE START AND STOP.

FIGURE 13. TWO-WIRE START AND STOP.

FULL AUTOMATIC STARTER, AUXILIARY DEVICES.

In order that an automatic starter may perform as a "full" automatic, the "start" and "stop" switch for hand operation is eliminated, and the voltage circuit of the shunt coils of the main contactor, or the current circuit of the pilot motor, is completed by auxiliary devices mechanically or electrically tied in with the duty load. Automatic devices similar to the principle of the three-wire start and stop button shown in Figure 12 may be operated by a device known as a sylphon, which contracts at low liquid or gas pressures, and closes the "start" circuit, and expands upon high pressure and stops the motor-driven pump or compressor. Other devices use the deflection of a bourdon gauge mechanism to close and open the auxiliary contacts at low and high pressures. Many other forms of mechanical motion may be employed to operate the auxiliary contacts such as bi-metallic metals, mercury level switches, levers operated by floats in tanks, time clocks, low voltage relays, etc. In Naval service, however, due to the fact that practically all full automatic starters are of small capacity, and usually operate on compound wound motors, step starters and auxiliary circuits are omitted, and the starter is operated as a line switch by means of quick acting levers and toggles.

STARTING CHARACTERISTICS.

The starting characteristic of a motor in terms of line ampères is the usual criterion of the successful design and application of the starter for the particular motor and installation. The torque on the motor shaft resulting from the line ampères, excepting the first current inrush, follows very closely the line current characteristics. Due to the fact that the field strength of the motor builds up relatively slowly compared with the current inrush to the armature, the torque builds up relatively slowly even at high first peak current. From there on, regardless of the number of succeeding peaks due to steps of starting,

the torque rises and falls practically proportionally to the line current. In order to exactly portray the effect on the line current at different stages of starting, the use of an instrument known as the oscillograph is necessary. This instrument, when operated to give the instantaneous values of current over a cycle of starting, furnishes a permanent photographic record of the current characteristics of which an analysis can be made to determine the correctness of design or adjustment of the starter for a particular application. Figures 15 to 22 are typical theoretical oscillograph curves and reproductions of actual oscillograph curves, showing some of the possible combinations and variations of results under various conditions. It is obvious from a study of the general characteristics that designers should give careful consideration to the problems involved in order that ideal starting conditions may be obtained.

EFFECT OF CHANGING THE VALUE OF STARTING RESISTANCE.

Starting characteristics showing the relative effect of changing the amount of starting resistance, are interesting and instructive. In order to readily calculate the amount of effective resistance in the motor line circuit, it is the practice of some engineers to speak of motors in terms of per cent resistance. Commercial motors vary in resistance from about 10 per cent for a small motor, to about 5 per cent for a large motor. For example, a motor rated at 200 volts 100 ampères full load current, is equivalent to a 2 ohm circuit. A 7 per cent motor would have a resistance of 7 per cent of 2 ohms or 0.14 ohms. If it is desired that the first peak current on starting be limited to 200 per cent full load current, the total resistance should be 1 ohm or 50 per cent. The external resistor would then be 50 per cent minus 7 per cent or 43 per cent = 0.86 ohms. If it is desired that the first peak be limited to 500 per cent full load current, the total resistance should be 0.4 ohms or 20 per cent. The external resistor would then be 20 per cent minus

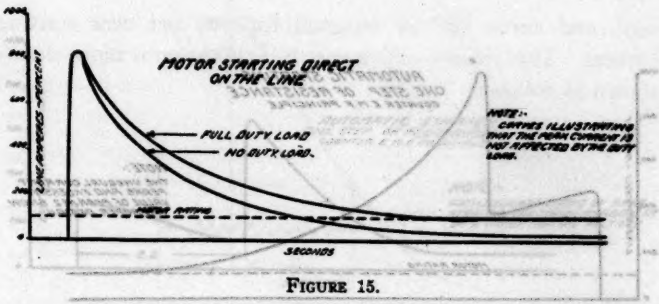

MOTOR STARTING DIRECT ON THE LINE
FULL DUTY LOAD
NO DUTY LOAD.
MOTOR RATING
SECONDS
NOTE:- CURVES ILLUSTRATING THAT THE PEAK CURRENT IS NOT AFFECTED BY THE DUTY LOAD.

FIGURE 15.

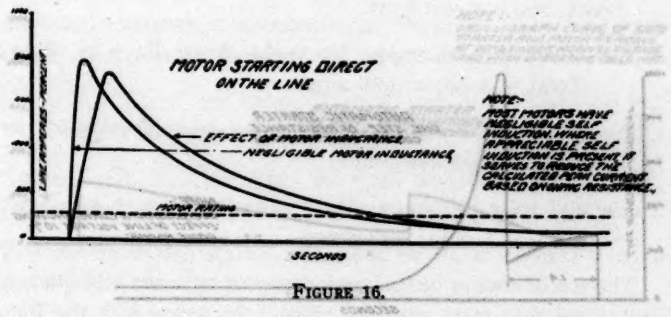

MOTOR STARTING DIRECT ON THE LINE
EFFECT OF MOTOR INDUCTANCE
NEGLIGIBLE MOTOR INDUCTANCE
MOTOR RATING
SECONDS
NOTE:- MOST MOTORS HAVE APPRECIABLE SELF INDUCTION, WHERE APPRECIABLE SELF INDUCTION IS PRESENT, IT SERVES TO REDUCE THE CALCULATED PEAK CURRENT BASED ON OHMIC RESISTANCE.

FIGURE 16.

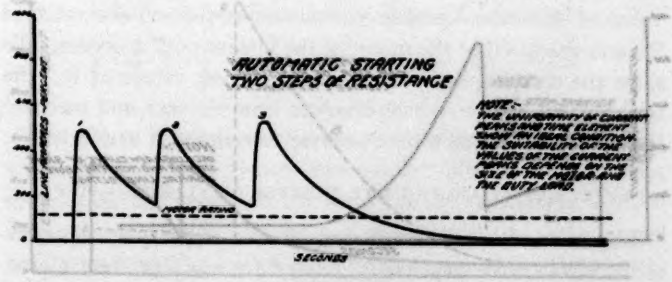

AUTOMATIC STARTING TWO STEPS OF RESISTANCE
MOTOR RATING
SECONDS
NOTE:- THE UNIFORMITY OF CURRENT PEAKS AND TIME ELEMENT SHOW IDEAL CONDITION. THE SUITABILITY OF THE VALUES OF THE CURRENT PEAKS DEPENDS ON THE SIZE OF THE MOTOR AND THE DUTY LOAD.

FIGURE 17.

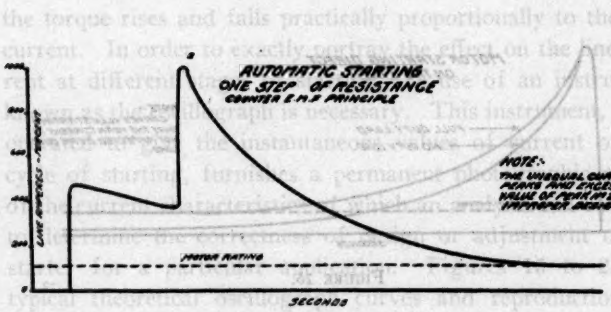


FIGURE 18.

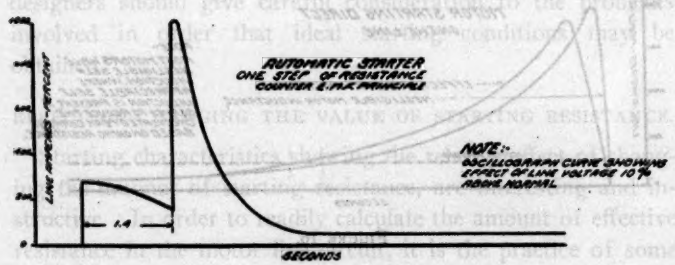


FIGURE 19.

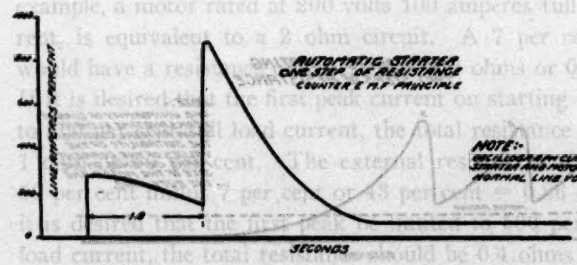


FIGURE 20.

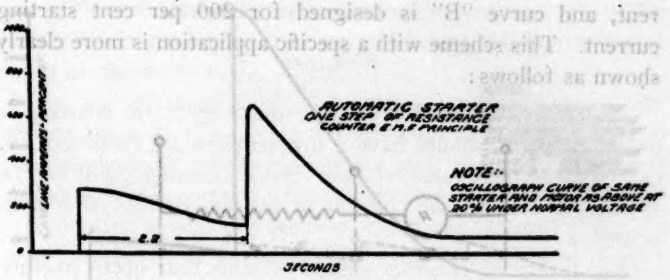


FIGURE 21.

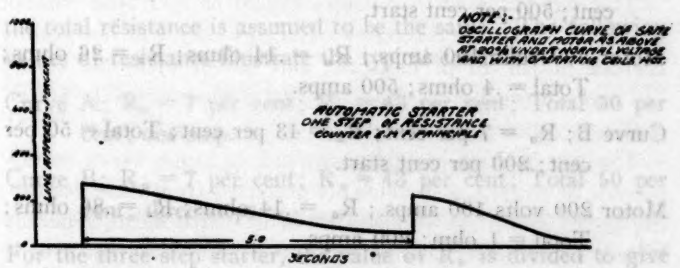


FIGURE 22.

7 per cent or 13 per cent = 0.26 ohms. Figure 23 shows the peak currents of a one step starter plotted for any 7 per cent motor. Curve "A" is designed for 500 per cent starting cur-

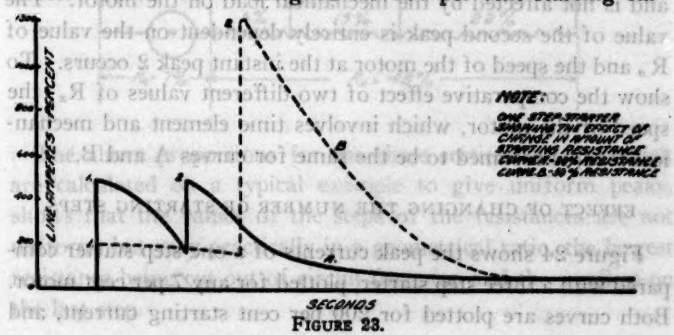
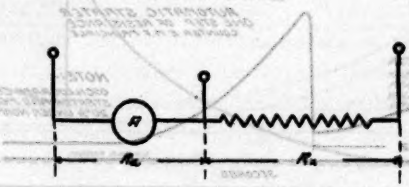


FIGURE 23.

rent, and curve "B" is designed for 200 per cent starting current. This scheme with a specific application is more clearly shown as follows:



Curve A; R_a = 7 per cent; R_x = 13 per cent. Total = 20 per cent; 500 per cent start.

Motor 200 volts 100 amps.; R_a = .14 ohms; R_x = .26 ohms; Total = .4 ohms; 500 amps.

Curve B; R_a = 7 per cent; R_x = 43 per cent; Total = 50 per cent; 200 per cent start.

Motor 200 volts 100 amps.; R_a = .14 ohms; R_x = .86 ohms; Total = 1 ohm; 200 amps.

There will always be two peak currents on a one step starter, one at the first start with the resistor in series with the line, and the other when the line is applied direct. The value of the first peak is entirely dependent on the values of R_a and R_x, and is not affected by the mechanical load on the motor. The value of the second peak is entirely dependent on the value of R_a and the speed of the motor at the instant peak 2 occurs. To show the comparative effect of two different values of R_x, the speed of the motor, which involves time element and mechanical load, is assumed to be the same for curves A and B.

EFFECT OF CHANGING THE NUMBER OF STARTING STEPS.

Figure 24 shows the peak currents of a one step starter compared with a three step starter, plotted for any 7 per cent motor. Both curves are plotted for 200 per cent starting current, and

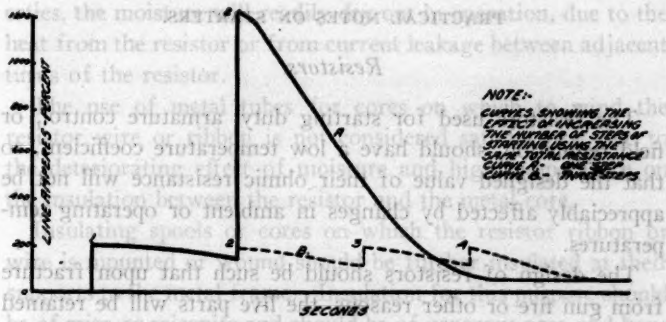

LINE AMPERES - PERCENT
SECONDS
NOTE:-
CURVES SHOWING THE EFFECT OF INCREASING THE NUMBER OF STEPS OF STARTING USING THE SAME TOTAL RESISTANCE
CURVE A.- ONE STEP
CURVE B.- THREE STEPS

FIGURE 24.

the total resistance is assumed to be the same. The following values of resistance illustrate the typical condition:

Curve A; $R_a = 7$ per cent; $R_z = 43$ per cent; Total 50 per cent; one step.

Curve B; $R_a = 7$ per cent; $R_z = 43$ per cent; Total 50 per cent; three steps.

For the three step starter, the value of R_z is divided to give equal peaks, into the following proportion:

The above proportions for the three sections of R_z which are calculated on a typical example to give uniform peaks, shows that the values of the steps of the resistances are not uniform, but vary practically in a geometrical ratio, the largest resistance being cut out of circuit first step and the smallest on the last step.

PRACTICAL NOTES ON STARTERS.

Resistors.

The resistors used for starting duty, armature control, or field regulation, should have a low temperature coefficient, so that the designed value of their ohmic resistance will not be appreciably affected by changes in ambient or operating temperatures.

The design of resistors should be such that upon fracture from gun fire or other reasons, the live parts will be retained in position so that short circuits or grounds are avoided.

The material used in resistors should be such that without the use of artificial protective coverings they will not readily corrode in salt air or moisture, and should withstand cherry red temperature for considerable time without appreciable oxidation or other deterioration.

The use of wire or ribbon wound resistors which readily corrode or oxidize, and which are protected from these destroying agents by refractory materials or insulated metal, generally have not been found satisfactory. Refractory materials usually have different coefficients of expansion than the resistors, which results in flaking or cracking. Metal clad resistors are invariably insulated from the covering by mica which flakes and crumbles after repeated cycles of heating and cooling.

Cast grid resistors of ferrous material of malleable construction and of slow oxidizing and corroding properties have been found satisfactory for large current ratings, if properly supported to withstand shock.

Resistors should be supported in an efficient manner so that they will not sag or bend at a cherry red temperature, thereby causing short circuits.

Insulating spools or cores on which the resistor wire or ribbon is wound, in order to withstand high temperature without deterioration, are usually of refractory materials. Such materials are hygroscopic, but due to their non-burning prop-

erties, the moisture will readily dry out in operation, due to the heat from the resistor or from current leakage between adjacent turns of the resistor.

The use of metal tubes for cores on which to wind the resistor wire or ribbon is not considered satisfactory, due to the deteriorating effect of moisture and high temperature on the insulation between the resistor and the metal core.

Insulating spools or cores on which the resistor ribbon or wire is mounted or wound should be further insulated at their supports to the metal frame. Insulation for this purpose should be of mica or micanite and should be of generous area and have ample creepage distance to ground so that current leakage through iron oxide, which usually forms at this point, will be avoided. Leakage at this point is a serious matter, whereas leakage between adjacent turns of the resistor is not a serious factor.

Contacts, Contactors and Switches.

In order to reduce the heating and power loss, the mechanical pressure at the point of contact is just as essential as the area of contact.

The use of grease or oil on sliding contacts is not detrimental except that it retains dust and grit. On the other hand, it prevents cutting of the metal, reduces friction, and lowers the power losses. It retains these characteristics for long periods of use and disuse. Contacts free from oil or grease rapidly oxidize and the power loss may increase to a comparatively large amount.

Magnetic blowout coils are very effective in suppressing arcing. They should be incorporated in design of all contacts where the current interruption is destructive. Where auxiliary carbon contacts are used to break the current, generous air-gap distances or insulating barriers are required to prevent the carbon arc from holding or grounding on interrupted overloads.

All current making contacts should operate on a sliding or rocking principle. The sliding movement is efficient in cleaning the metal contact, and the rocking principle serves to present part of the contact area for the arcing, and part for carrying the running current.

Flexible copper pigtails should be used where movable contacts would otherwise depend on hinge pins, coiled springs or shunt sockets.

Lever knife blade type of switch should not be used to open a main current circuit over 50 ampères. The circuit should be broken by magnetic blowout contactors, or by auxiliary carbon break circuit breakers.

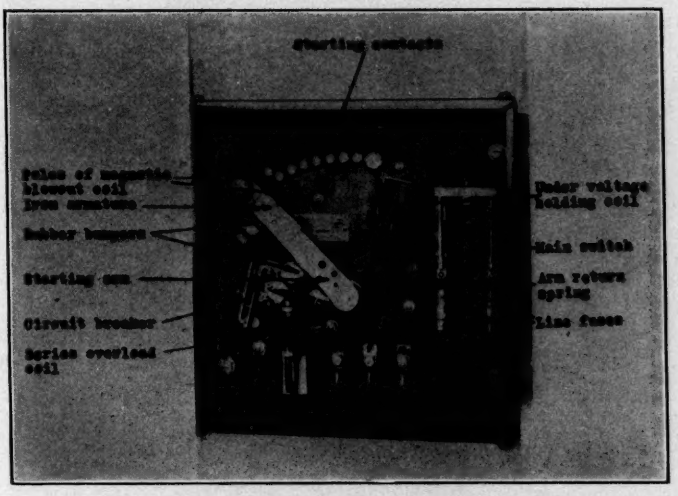
General.

The polarity of the line switch regardless of which side is positive or negative, has no effect on the proper operation of the starter or direction of rotation of the motor. In fact, schemes have been tried to periodically reverse the line polarity to equalize the pitting of the motor commutator under the negative brushes, and the blackening effect under the positive brushes.

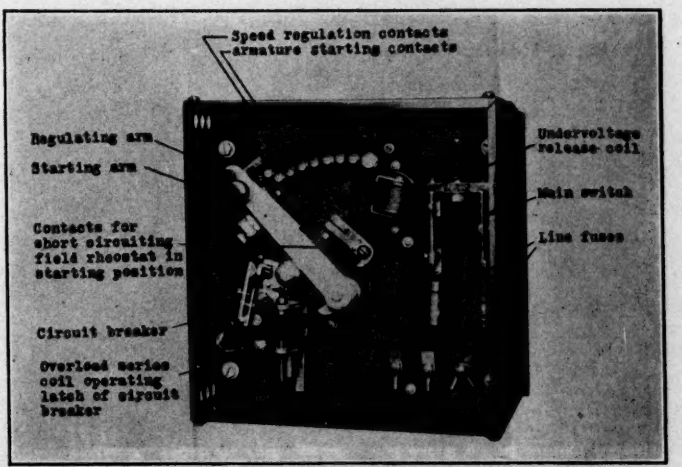
Shunt closing coils on automatic contactors should be designed so that the current in the coil is automatically reduced when the contactor is in the holding position. This arrangement gives the maximum pull with the minimum of size and power loss.

Shunt coils in which the function is affected by changes in line voltage should be copper wound and in series with a resistor of large percentage resistance of low temperature coefficient, so as to reduce the effect of temperature on the shunt coil. Designs using saturated cores are not affected to any extent by change in voltage or temperature.

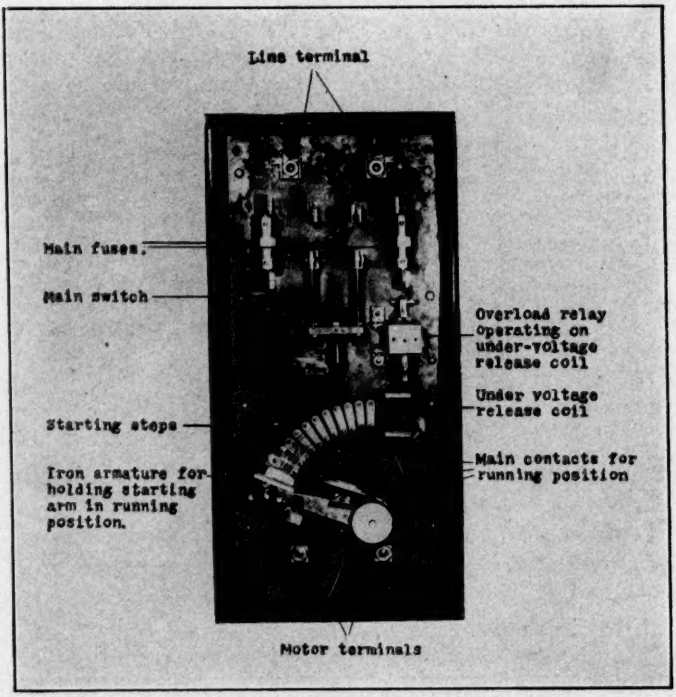
Manually operated controllers with shunt field speed control should be mechanically designed so that the motor cannot be started except on full shunt field. On automatic controllers,



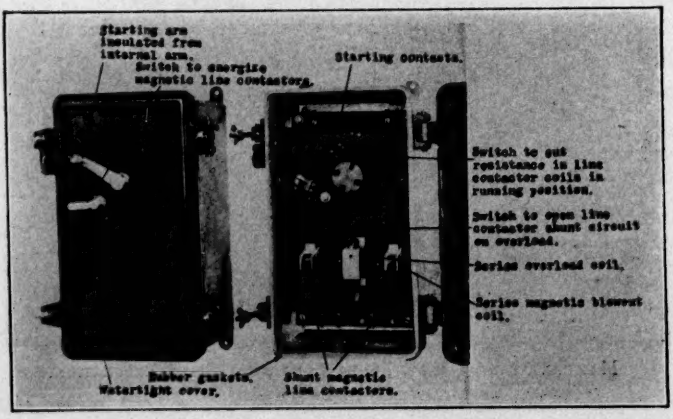
MANUAL STARTER, FACE PLATE TYPE.



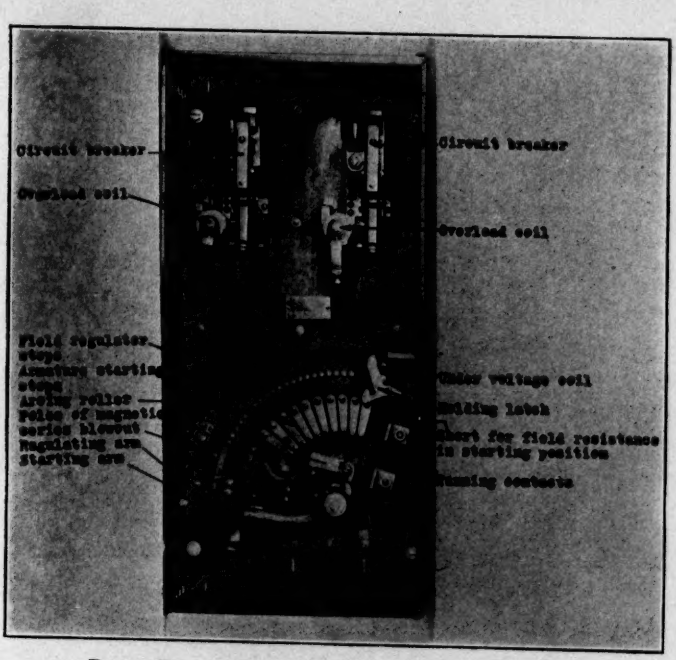
MANUAL STARTER, FACE PLATE TYPE WITH SPEED REGULATION.



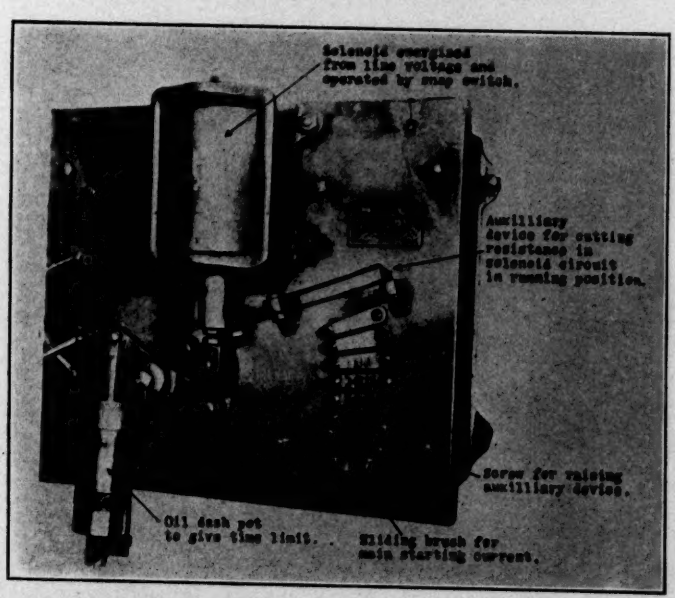
FACE PLATE, MANUAL TYPE, STARTER FOR 50 H. P. MOTOR, INFREQUENT DUTY.


Starting arm insulated from internal arm.
Switch to energize magnetic line contactors.
Starting contacts.
Switch to cut resistance in line contactor coils in running position.
Switch to open line contactor shunt circuit on overload.
Series overload coil.
Series magnetic blowout coil.
Rubber gaskets.
Watertight cover.
Shunt magnetic line contactors.

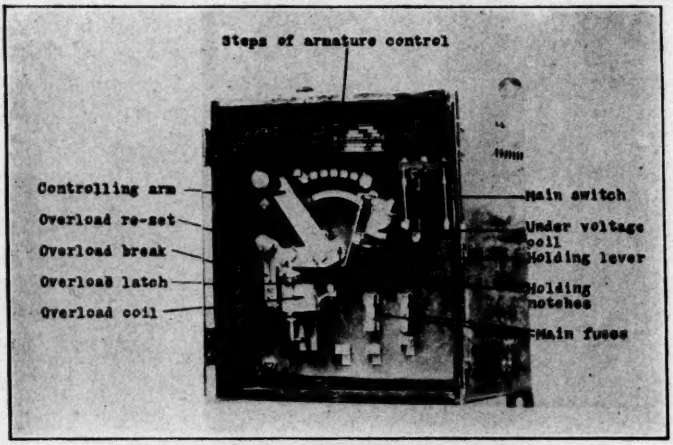
MANUAL STARTER, FACE PLATE TYPE WATERTIGHT, EXTERNALLY OPERATED.



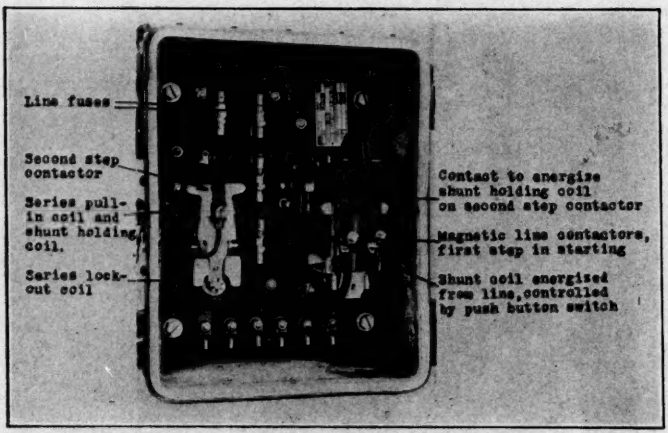
PANEL TYPE, MANUAL STARTER AND SPEED REGULATOR.


Solenoid energized from line voltage and operated by snap switch.
Auxiliary device for cutting resistance in solenoid circuit in running position.
Screw for raising auxiliary device.
Oil dash pot to give time limit.
Sliding brush for main starting current.

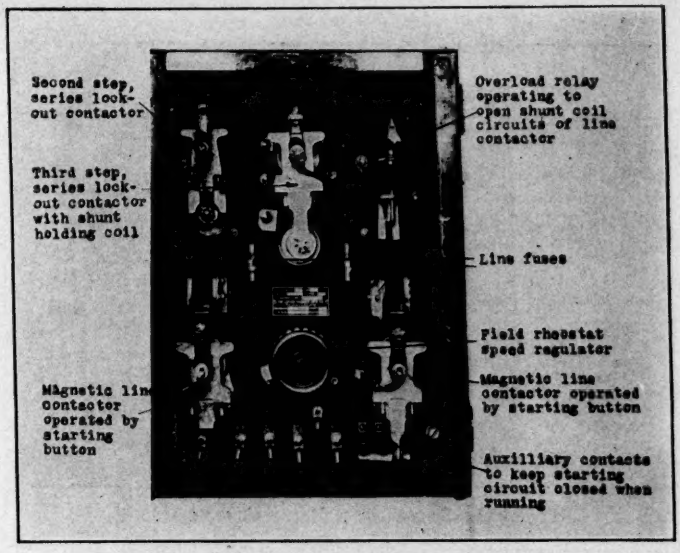
FACE PLATE, AUTOMATIC, TIME LIMIT, STARTER.



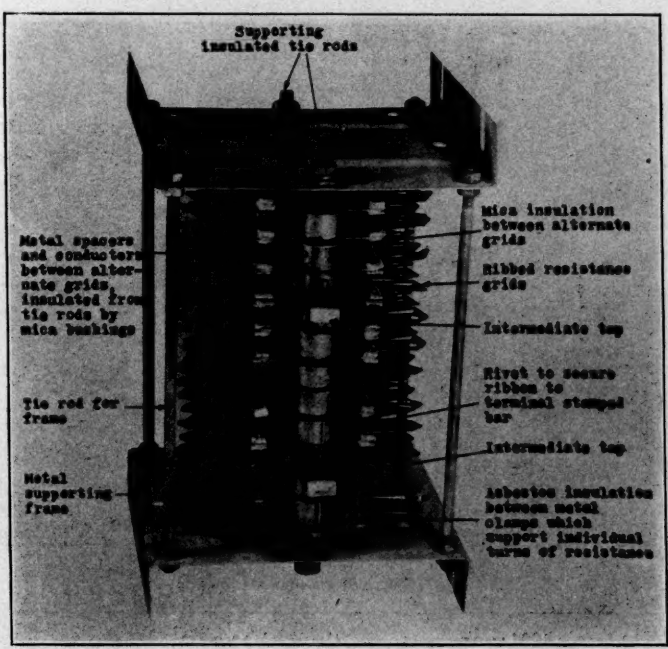
FAN REGULATOR, ARMATURE CONTROL.


Line fuses
Second step contactor
Series pull-in coil and shunt holding coil.
Series lock-out coil
Contact to energise shunt holding coil on second step contactor
Magnetic line contactors, first step in starting
Shunt coil energised from line, controlled by push button switch

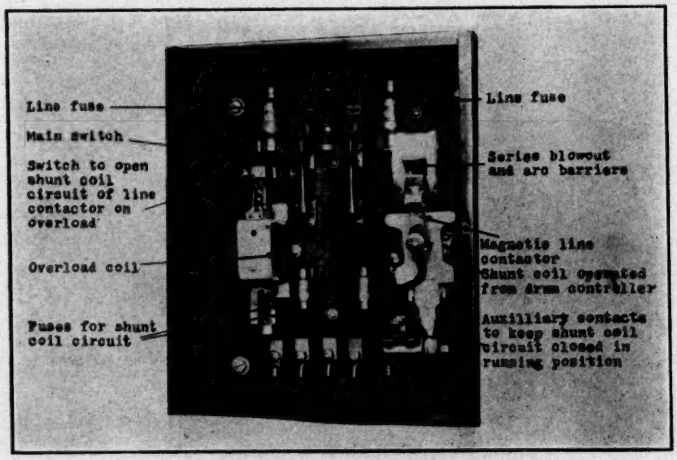
AUTOMATIC, SERIES LOCK-OUT TWO-STEP STARTER.


Second step, series lock-out contactor
Third step, series lock-out contactor with shunt holding coil
Magnetic line contactor operated by starting button
Overload relay operating to open shunt coil circuits of line contactor
Line fuses
Field rheostat speed regulator
Magnetic line contactor operated by starting button
Auxilliary contacts to keep starting circuit closed when running

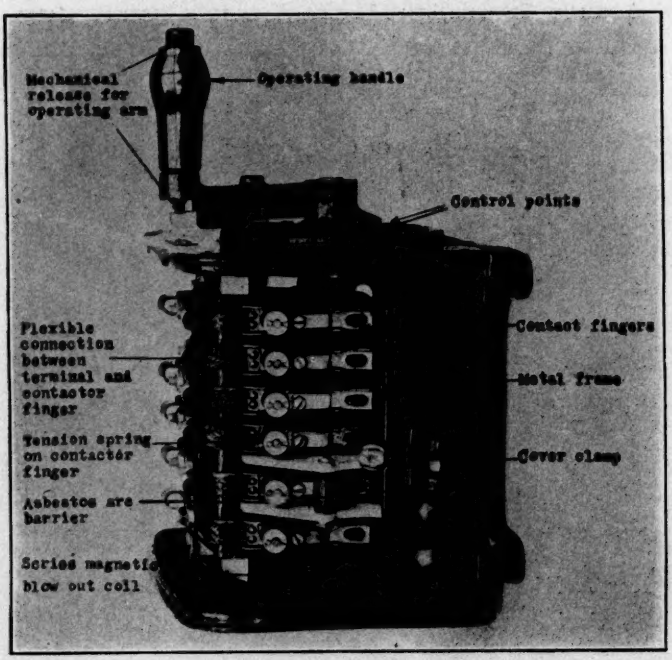
AUTOMATIC STARTER, SERIES LOCK-OUT TYPE WITH SPEED REGULATOR.


Supporting insulated tie rods
Metal spacers and conductors between alternate grids, insulated from tie rods by mica bushings
Tie rod for frame
Metal supporting frame
Mica insulation between alternate grids
Ribbed resistance grids
Intermediate tap
Rivet to secure ribbon to terminal stamped bar
Intermediate tap
Asbestos insulation between metal clamps which support individual turns of resistance

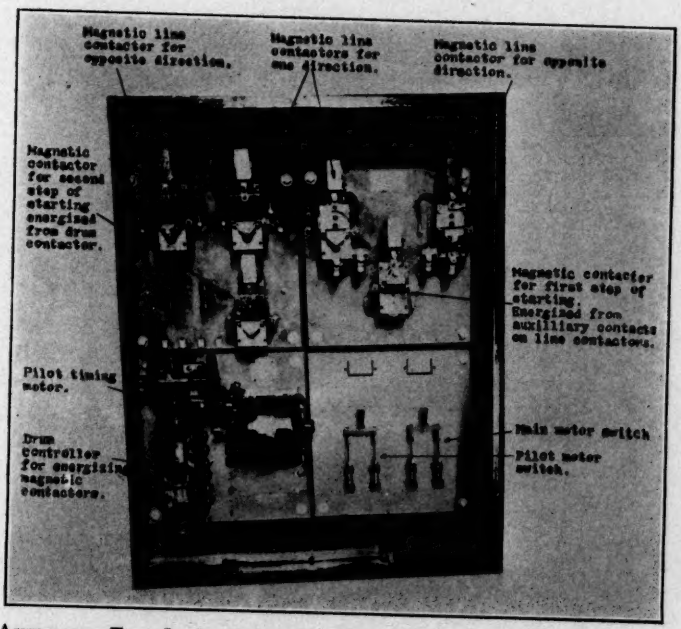
STARTING RESISTOR, STAMPED RIBBON NICKEL-CHROMIUM RESISTANCE UNITS.


Line fuse
Main switch
Switch to open shunt coil circuit of line contactor on overload
Overload coil
Fuses for shunt coil circuit
Line fuse
Series blowout and arc barriers
Magnetic line contactor
Shunt coil operated from drum controller
Auxilliary contacts to keep shunt coil circuit closed in running position

PROTECTIVE PANEL FOR A DRUM CONTROLLER.



MANUAL REVERSING DRUM CONTROLLER.


Magnetic line contactor for opposite direction.
Magnetic line contactors for one direction.
Magnetic line contactor for opposite direction.
Magnetic contactor for second step of starting energized from drum contactor.
Magnetic contactor for first step of starting. Energized from auxiliary contacts on line contactors.
Pilot timing motor.
Main motor switch
Drum controller for energizing magnetic contactors.
Pilot motor switch.

AUTOMATIC, TIME LIMIT, CONTACTOR TYPE, REVERSING, TWO STEP STARTER.

the regulating field rheostat should be automatically short circuited on starting and automatically cut in circuit to a predetermined setting when the armature current reduces to a safe amount.

Conclusion.

It would appear from the description in the foregoing paragraphs of the number of types and principles available for consideration in the starting and control of D. C. motors, that the selection of the most suitable combination for satisfactory results is somewhat involved. Although there may possibly be more than one general type or principle which will give equal results for a specific application, nevertheless, it is apparent that considerable study of the characteristics involved is necessary in order that the purchaser may be conversant with the characteristics of the various types on the market. For Naval service, it is believed as a guide to engineers who deal with the specification, selection and application of starters and controllers, the following general features are essential in Naval application:

Manual operated starters for frequent duty, not to exceed 4 horsepower.

Automatic starters to be of the current limit principle.

Upon shut down of motor from overload or low voltage, the starter to require manual re-set.

Excessive peak currents should be avoided at any stage of starting.

The effect of high or low line voltage to a reasonable degree should be negligible.

Ship vibration and shock of gun fire should not result in the starting or stopping of the motor.

Protection from the effect of moisture and corrosion should be carried to a high degree.

FUEL OIL VISCOSITY.*

By Lieut. Comdr. G. B. Vroom, U. S. Navy, Member.

A present want, and one that will be felt increasingly as heavier and more viscous oils are marketed, is a simple and accurate means of determining, recording and interpreting the viscosity curves of fuel oils. As an operating problem, the handling of viscous fuel oil, with respect to its fluidity, must be considered not only as regards the supply system to the atomizers, but also its handling from the tanks, or bottoms, to the service tank from which the supply to the atomizers is taken. It has been found that preheating of the more viscous oils, in the storage tanks, is necessary in order that the pumps will take a suction, particularly when the temperature of the oil is below 50 degrees F.

The two definite parts of the problem are:

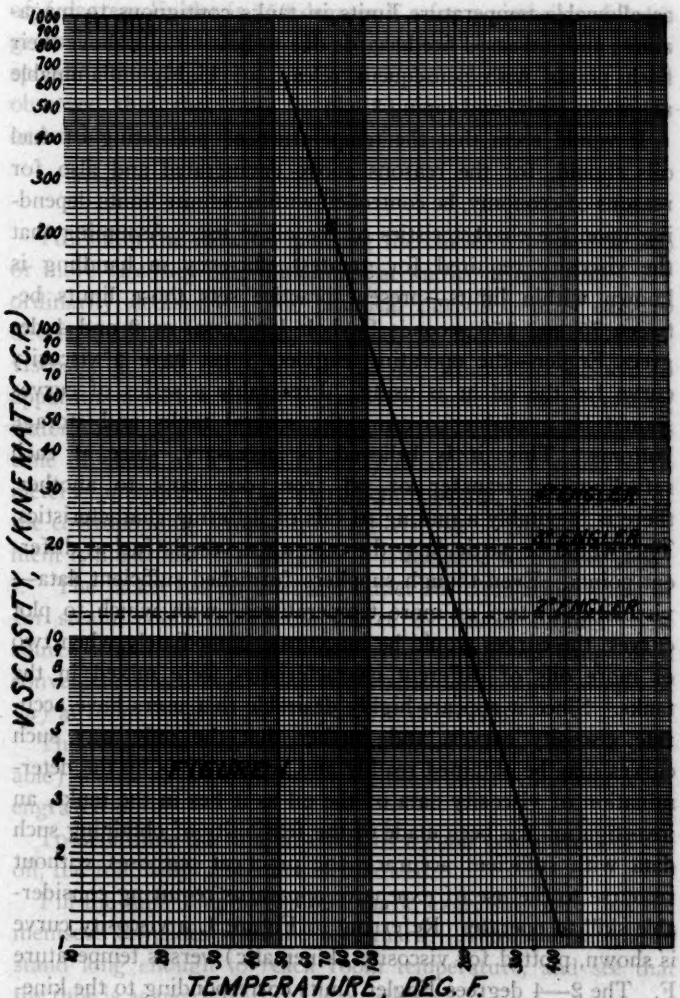
(*a*) Temperature to which oil in tanks or bottoms must be heated to permit ready handling.

(*b*) Temperature to which oil should be heated in the supply system to the atomizers.

There are many factors entering into the ultimate solution of these problems, about which little is known; these factors may be mechanical, in so far as they concern the design of the pumps, the fuel lines, the "hot oil recirculating" system, and the heaters; they may have to do with the characteristics of

* The purpose of this discussion is to demonstrate to the operating engineer, a practical application of certain known facts in petroleum technology, as a ready means of solving an ever present operating problem: *i.e.*, having a certain grade of fuel oil in the bottoms, what temperatures are necessary to reduce it to the prescribed Engler viscosities? It is well recognized by the author that the solution demonstrated will not produce laboratory exactness; on the other hand such a degree of refinement could hardly be utilized under operating conditions.

The use of the logarithmic scale in no way complicates the practicability of the methods demonstrated; logarithmic cross-section paper is readily procurable and, for that matter, can be easily constructed to any desired unit scale, by methods described in "Kent" and similar hand-books.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TEMPERATURE, DEG. F.

the oils procurable, and finally they may be collateral, such as allowable temperature limits in tanks contiguous to magazines; and increased fire hazards due to heating oils above their flash points with consequent release of highly inflammable vapors in tanks and piping.*

It would seem that the temperature of preheating of fuel oils should, for the reasons above enumerated and also for reasons of economy, be controlled within definite limits depending upon the characteristics of the oil in use. Assuming that the viscosity required for greatest efficiency in handling is known within limits—present practice sets those limits between 75 and 150 seconds Saybolt Universal or 2 and 4 degrees Engler—the operating engineer must have a viscosity curve for the oil he is using. Preferably a series of curves should be available covering the several tanks and storage bottoms. This, it is believed, is necessary, since at each refueling, or transfer of oil from one tank to another, there is inevitably a mixing of oils of differing characteristics. This would be particularly the case when vessels fuel at different points. Even though suppliers furnished sufficient data—viscosities at two or more temperatures—with which to plot curves, the curves would be inaccurate immediately when two or more oils of different characteristics were mixed in the tanks. Since it appears highly desirable that users have accurate viscosity curves, and further that the only way such curves may be obtained readily and accurately is by the determination of viscosity data of the oil as it lies in the tanks, an investigation has been made of the feasibility of obtaining such data with sufficient accuracy for practical purposes without recourse to the use of expensive apparatus requiring considerable skill to operate. Referring to Figure 1, a viscosity curve is shown, plotted for viscosity (kinematic) versus temperature F. The 2—4 degrees Engler limits corresponding to the kinematic viscosity scale are shown in dotted lines. It is evident from this Figure that, for the given oil represented by the

* The Fire Hazards of Fuel Oil—"Marine Engineering and Shipping Age," May, 1925.

curve, the temperature F. necessary to give the oil a viscosity of 4 degrees Engler is 150 degrees F.; for a viscosity of 2 degrees Engler, 200 degrees F.; any other viscosity and corresponding temperature can of course be selected. It is to be observed that the curve is plotted to logarithmic scale, in order that it shall appear as a straight line.

II.

To obtain the data for plotting the curve in Figure 1, two or more viscosity determinations, at known temperatures, are ordinarily necessary. For practical purposes, it is easy to devise a simple apparatus, of glass, by means of which the viscosity at two temperatures can be determined. Such an apparatus would be simply a glass tube, jacketed for circulating water, having a capillary orifice, and so calibrated that the time of efflux of a given quantity of oil—say 5 cubic centimeters—multiplied by the instrument constant, would give, for the corresponding temperature, the viscosity. Such an instrument is illustrated in Figure 2. T is the tube with glass jacket *j*. The jacket has an inlet and outlet for circulating water or for steam, connected to the source by rubber tubing. The source would be a small heater using an alcohol lamp; or a convenient steam line. The two viscosities obtained would be (*a*) at room temperature, (*b*) at 212 degrees F.

The receptacle into which the oil flows (a test tube is suitable) should have a mark, showing the 5 cubic centimeter level, engraved on it.

To determine the time of efflux of 5 cubic centimeters of the oil, the following procedure is followed.

Fill the tube T up to the zero mark, exercising care that the meniscus coincides with the zero mark. Allow the sample to stand long enough to reach room temperature, and see that the tube is as nearly vertical as possible.

Remove stopper from the orifice, simultaneously starting the stop-watch. When the 5 cubic centimeter level marked on the

receptacle is reached, stop the watch. The elapsed interval is the time of efflux.

Then $V_k = At$

Where A is the tube calibration constant, and t the time of efflux.

V_k, plotted to the scale of Figure 1, gives one point on the curve.

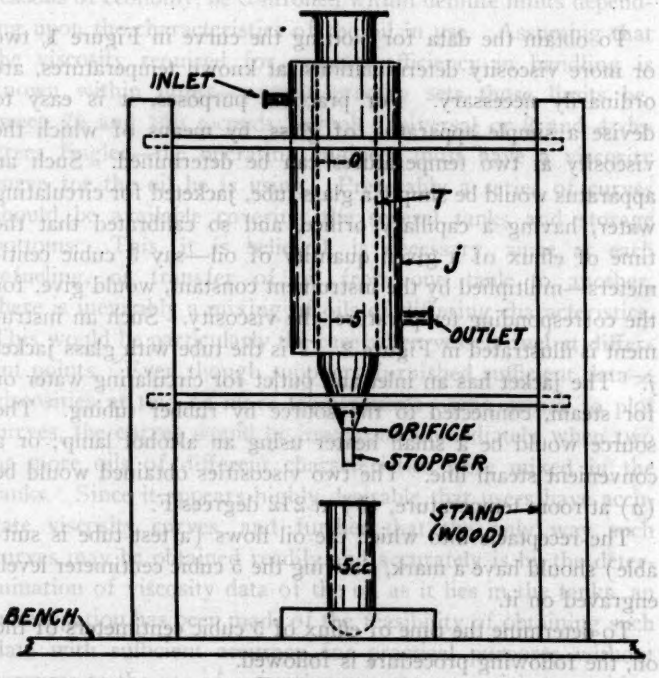


FIGURE 2.

The procedure for obtaining the second point is the same, except that steam or boiling water is circulated through the jacket, and the sample brought to constant temperature, which should be measured by thermometer.

(NOTE 1: A more obvious procedure for measuring the time of efflux would be to fill the tube *above* the zero mark, and time with a stop-watch the passing of the meniscus at the zero and 5 cubic centimeter marks on the tube T. This can be done in the case of lubricating oils; because the oils are translucent; but cannot be done with fuel oils because these are opaque, and, further cling to and completely obscure the walls of the tube, so that the meniscus is not visible, once the oil starts to flow. Care should be taken to remove the stopper quickly, and away from the stream of oil flowing from the orifice.)

(NOTE 2: The above described apparatus is simple in construction, and cheap to make. While the calibration constant A, for the individual tube, should preferably be determined in a laboratory, it is not difficult to determine it, if the operator can obtain a few samples of oils whose viscosity curves are known.)

t = time of efflux of 5 cubic centimeters of oil of known viscosity.

For the given room temperature, the known viscosity (Saybolt Universal) of the oil is converted to Viscosity (kinematic) according to the equation

$$V_k = 0.22\ V_s - \frac{180}{V_s}$$

where V_s is the Viscosity (Saybolt Universal) in seconds. Then, since for the tube, $V_k = At$

$$A = \frac{V_k}{t}.$$

The writer has had no difficulty in constructing and calibrating apparatus as described above. Subsequent tests of viscosity of a large number of lubricating and fuel oils, proved that the accuracy of results is well within limits necessary for practical use; comparison of curves so obtained with curves of the same oils obtained with laboratory apparatus showed an agreement within 1 per cent.

III.

It has been observed that the viscosity curves of petroleum, when plotted to a logarithmic scale, are to all intents and purposes, straight line functions*. Wilson and Balke† have shown that specific gravity curves are likewise straight line functions; that the linear relation does not extend to high temperatures, and that, *with the exception of the heavier oils,* it is not safe to use (the linear relation) beyond 200 degrees F.‡

It is doubtful whether viscosity curves obtained and plotted as described in Sections I and II, or for that matter, from any series of determinations made in the laboratory at temperatures below 200 degrees F. are accurate above 200 degrees F. particularly for blended oils. A great majority of the oils now marketed, however, have characteristics such that their viscosities Engler (2 degrees—4 degrees) have corresponding temperatures below 200 degrees F.; therefore, for practical use, curves may be plotted, and temperatures corresponding to desired "operating" viscosities taken out with a practical degree of accuracy.

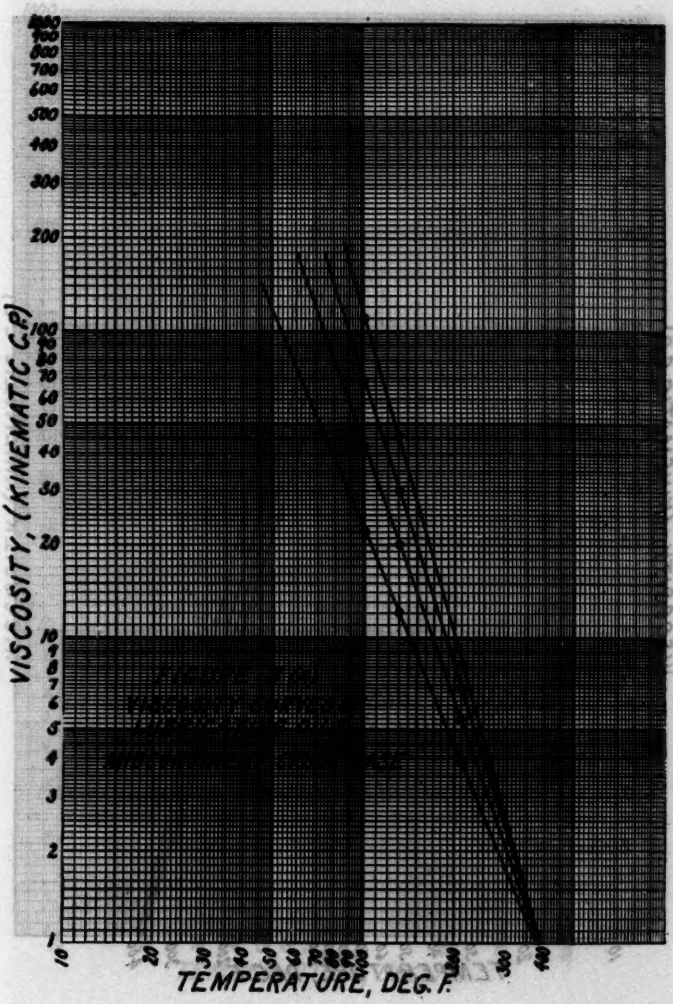
Not only do viscosity curves become straight line functions, when plotted to logarithmic scale, but such straight line functions for a wide variety of oils meet in a point, or within a small area, which, for want of a better definition, may be called "the origin of curves." This fact, observed by various investigators and in particular by Deane and Lane,* was independently observed by the author.

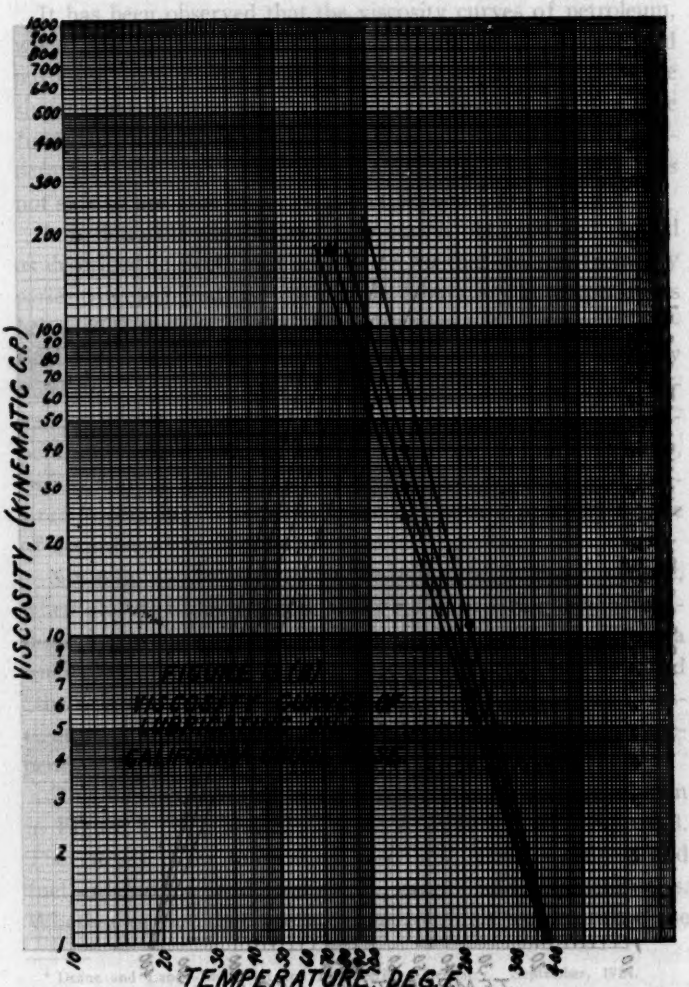
This fact is illustrated by the series of viscosity curves shown in Figure 3 (*a, b* and *c*); Figure 4; Figure 5, and Figure 6. The curves cover lubricating fractions; various crudes, and fuel oils, of a wide range of origin and characteristics. Whether or not they are accurate for high temperatures, the

* Deane and Lane—Industrial and Engineering Chemistry. September, 1924.

† "Mechanical Engineering," July, 1925, p. 537.

‡ $Vk = \frac{\text{Absolute Viscosity (c. p.)}}{\text{Specific Gravity}}$


VISCOSITY, (KINEMATIC C.P.)
TEMPERATURE, DEG. F.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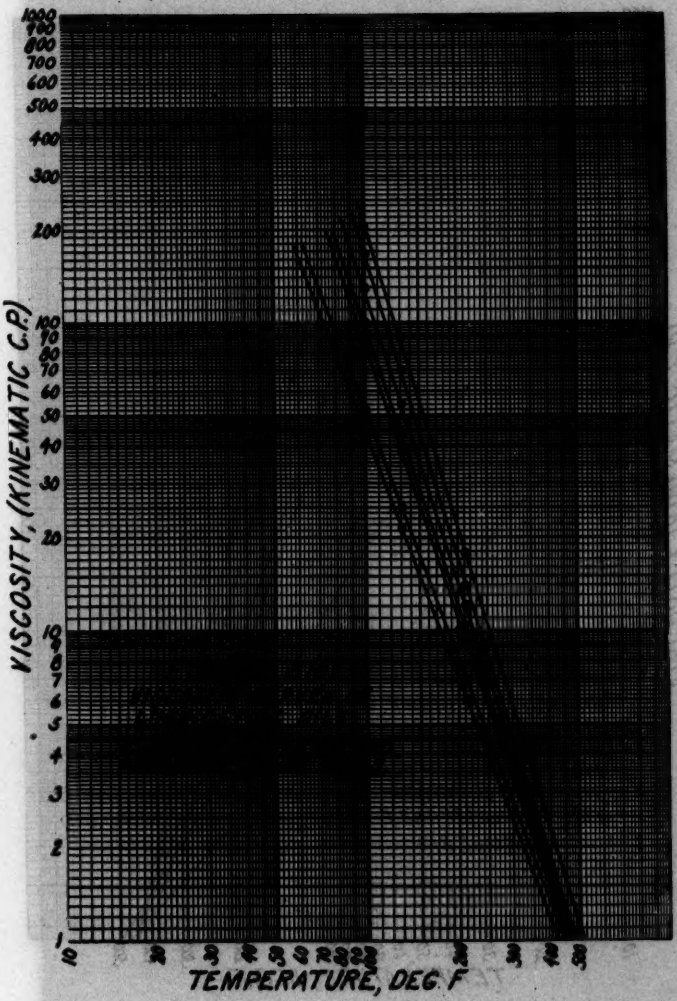
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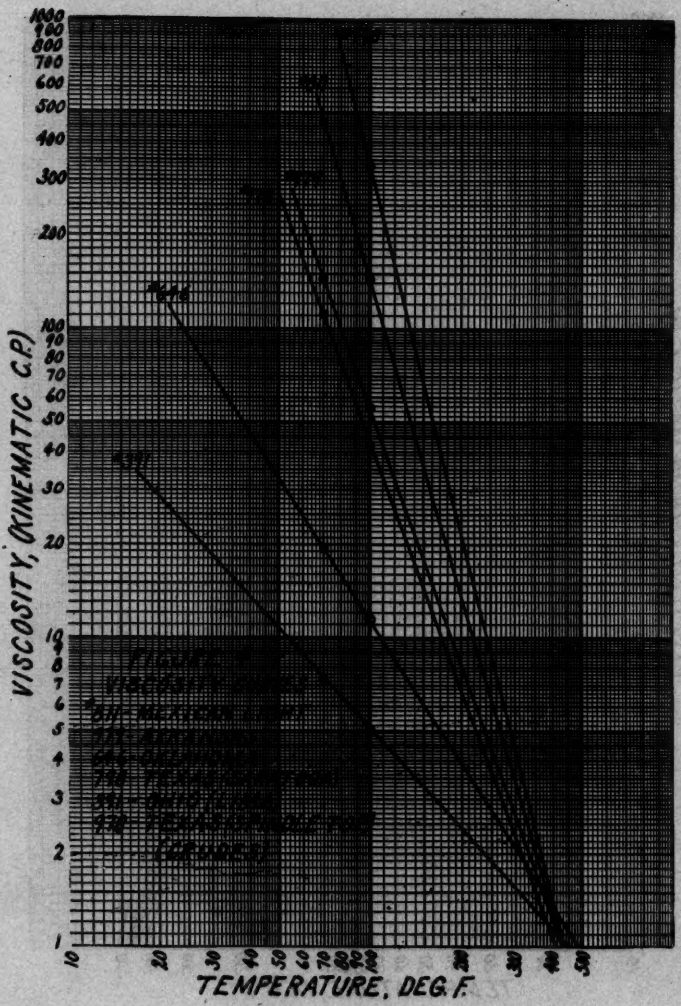
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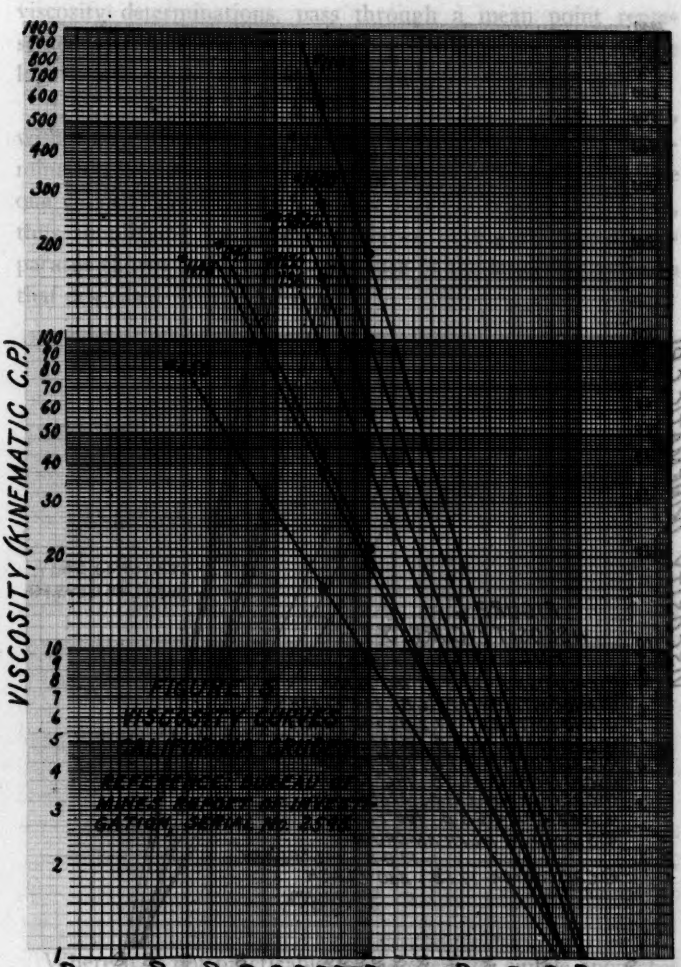
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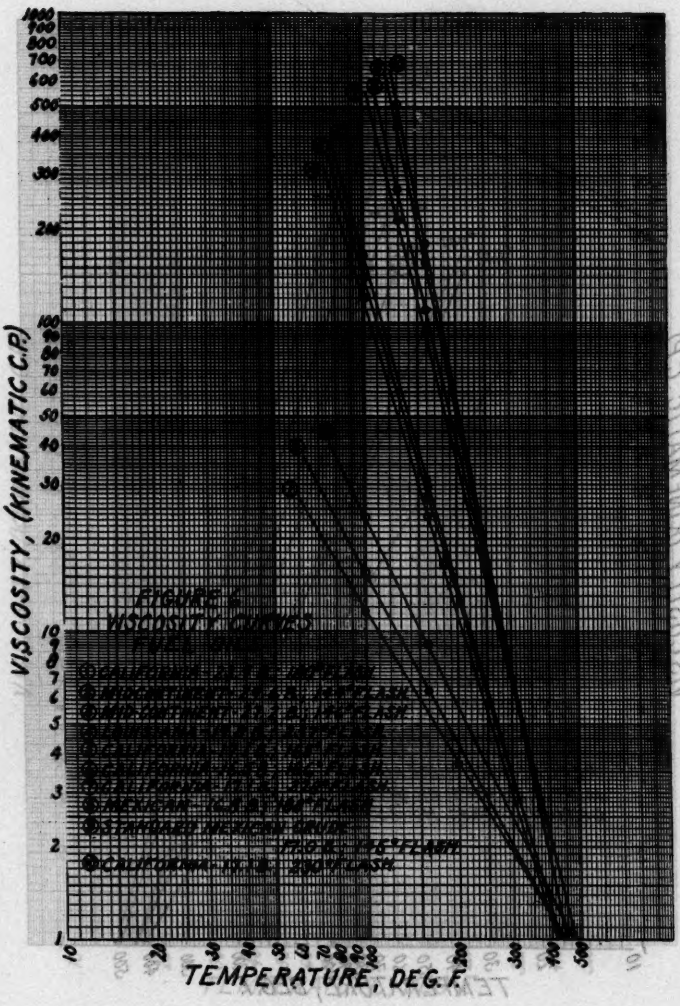
TEMPERATURE, DEG. F.


VISCOSITY, (KINEMATIC C.P.)
TEMPERATURE, DEG. F.

curves (or their continuations), if plotted for two or more viscosity determinations, pass through a mean point represented by a temperature of 450 degrees F. to the scale of the logarithmic paper used in these Figures.

It would appear, then, that the "mean point" may be used, with practical accuracy, in conjunction with *one* viscosity determination, to determine the viscosity curve of a given oil. The one viscosity determination may be made at room temperature, thus simplifying the apparatus shown in Figure 2 to an apparatus shown in Figure 7. The use of the latter is similar to that described for the former.

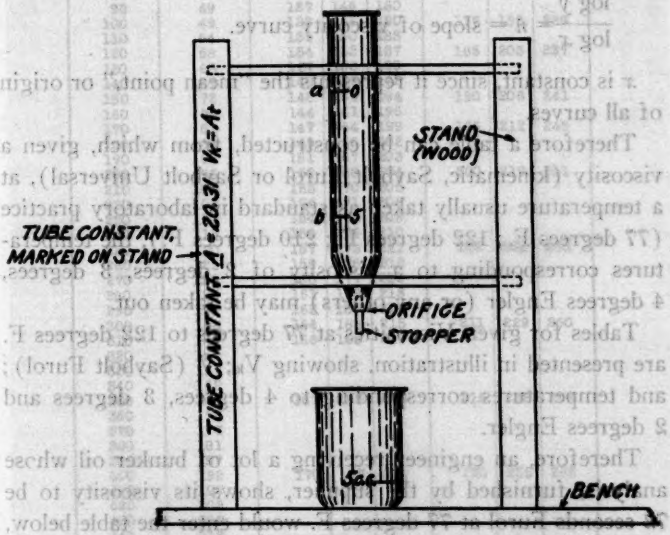


FIGURE 7

Where it is not practicable to use even the simple apparatus and methods described thus far, an operating engineer can determine with considerable accuracy something of the viscosity characteristics of oil with which he is supplied, by

referring the viscosity data usually supplied by the producer with a shipment of fuel oil, either cargo or bulk, to tabulated data based on the assumption deduced from Figures 3 to 6, that viscosity curves may be plotted from a "mean point" and a single viscosity determination.

The equation of a straight line function to logarithmic scale, is of the form

$$y = x^n$$

or

$$\log y = n \log x$$

from which

$$\frac{\log y}{\log x} = n = \text{slope of viscosity curve.}$$

x is constant, since it represents the "mean point," or origin of all curves.

Therefore a table can be constructed, from which, given a viscosity (kinematic, Saybolt Furol or Saybolt Universal), at a temperature usually taken as standard in laboratory practice (77 degrees F.; 122 degrees F.; 210 degrees F.), the temperatures corresponding to a viscosity of 2 degrees, 3 degrees, 4 degrees Engler (or any others) may be taken out.

Tables for given Viscosities at 77 degrees to 122 degrees F. are presented in illustration, showing V_k; V (Saybolt Furol); and temperatures corresponding to 4 degrees, 3 degrees and 2 degrees Engler.

Therefore, an engineer receiving a lot of bunker oil whose analysis, furnished by the supplier, shows its viscosity to be 72 seconds Furol at 77 degrees F. would enter the table below, column 2, and read off, in columns 3, 4 and 5, respectively, 142 degrees F. for 4 degrees Engler; 159 degrees F. for 3 degrees Engler and 194 degrees F. for 2 degrees Engler. These are the approximate temperatures required to be used in the heating system, to reduce the oil to the specified viscosities Engler.

FUEL OIL VISCOSITY - TEMPERATURE TABLE.

Viscosity Kinematic c.s.	Viscosity Saybolt Furol (seconds)	Given Viscosity at 77°F., find corresponding temperatures for			Given Viscosity at 122°F., find corresponding temperatures for		
		4°E	5°E	8°E	4°E	5°E	8°E
70	36	115	133	168			
72	37	117	135	170			
74	38	118	136	171			
76	39	119	137	171			
78	40	120	138	172			
80	41	120	139	173			
82	42	121	140	174			
84	42	122	140	174			
86	43	123	141	176			
88	44	124	141	177			
90	45	125	142	178			
92	46	125	142	179			
94	47	126	145	179			
96	48	126	144	179			
98	49	127	146	180			
100	49	128	146	180	177	185	229
110	54	131	150	185			
120	58	134	152	187	185	203	237
130	63	137	155	190			
140	67	140	157	192			
150	72	142	159	194	190	208	241
160	76	144	161	196			
170	81	147	164	199	195	212	245
180	86	148	166	202			
190	90	151	167	203			
200	95	152	168	204	200	219	251
210	100	153	170	205			
220	105	154	171	206			
230	110	154	172	208			
240	115	156	174	210			
250	120	157	175	211	206	226	255
260	125	159	176	212			
270	130	160	177	212			
280	136	161	178	213			
290	139	163	180	214			
300	145	164	182	215	211	229	260
310	147	166	185	216			
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350	167	167	185	218	215	252	262
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400	192	170	187	220	220	235	265
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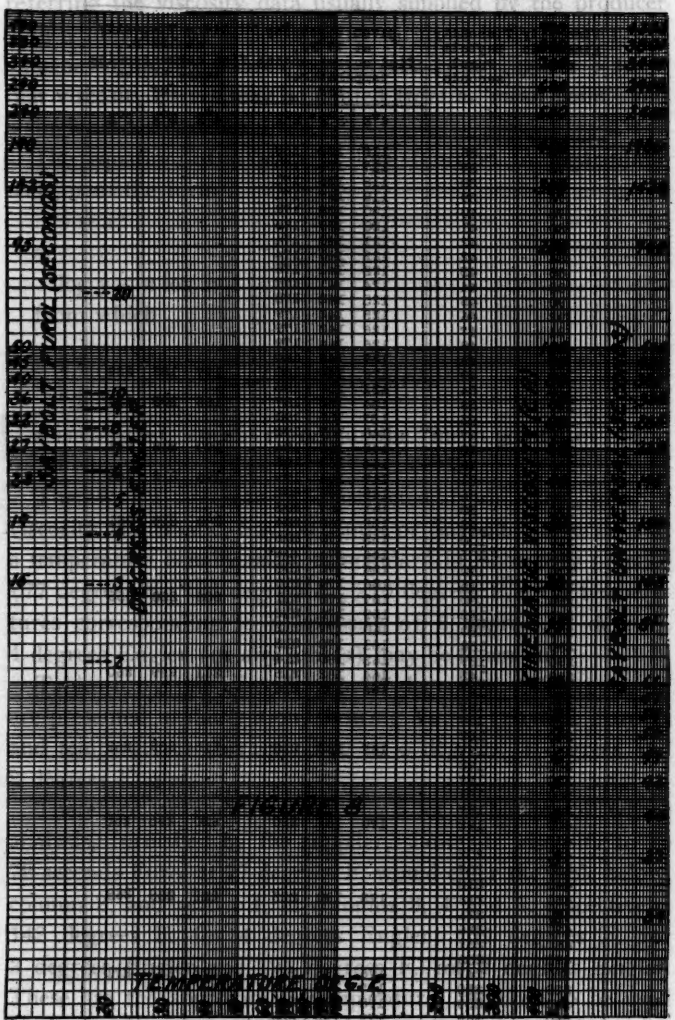
440	212						
450	216	173	190	223	222	238	268
460	221						
470	226						
480	231						
490	236						
500	241	176	195	226	225	241	271
520	250	177	198	228			
540	259	178	196	230			
560	269	179	196	230			
580	279	180	197	231			
600	289	181	198	232	229	245	275
700	335	184	202	234			
800	385	187	204	237			
900	430	191	208	241			
1000	480	192	208	241			
1100	525	195	212	243			
1200	575	197	214	245			



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For Naval use particularly, the simplest procedure of all would be the use of standard plotting sheets, similar to Figure 8. This sheet, on logarithmic paper, shows the several standard viscosity scales, versus Temperature F. Point "A" is the "mean point," or origin of curves. Therefore, a curve could be directly plotted from one viscosity (furnished by the supplier of the oil, or determined by the apparatus of Figure 7) and Point "A"; or from two viscosity determinations found with the apparatus of Figure 2.

A viscosity-temperature diagram such as that shown in Figure 8, can be mounted on heavy cardboard; the addition of a celluloid pointer, pivoted at point "A" and having a heavy line engraved or drawn on it, so that the line passes through the point "A", makes a handy instrument for quick use. The necessity of drawing the viscosity curve through the plotted point, for one temperature and corresponding viscosity, and the point "A", is thus obviated.

A PROPOSED METHOD FOR TESTING OILS AND RECLAIMING USED GAS ENGINE OILS.

By Lieut. Commander Z. W. Wicks, U. S. N., Member, and Lieutenant R. F. McCall, U. S. N.

In recent years some attention has been paid to recovery of used gas engine oils. In most instances these efforts have been put forth by individuals on a small scale. These attempts have been successful in that oil has been recovered, but they have paralleled to a great extent present methods of refining lubricating oils, with all of the attendant difficulties as well as a few additional ones.

Present practice in oil refining consists of distillation at atmospheric pressure, using a steam jet in the still to reduce the partial pressure of the oil vapor. This will reduce the boiling temperature below that obtained when distilling without steam, but the reduction is not sufficient to totally eliminate disintegration of the oils. As a result, the distillate must be "sweetened" to remove the undesirable products formed during exposure to the high temperature. Practice today includes the use of sulphuric acid and caustic soda consecutively. The resultant refined product shows no indications, according to the usual tests, that the oil is anything but 100 per cent perfect.

Conditions at the Helium Production Plant were such that an investigation of the oil situation was necessary. Consumption of oil in fourteen (14) natural gas 250 H.P., Bruce-Macbeth, engines was high and the used oil showed an unusually high percentage of soft carbonaceous residue. In an effort to determine the real character of the oil used, samples were distilled under 1/10 of a millimeter mercury pressure, using ordinary laboratory equipment. Sixteen different oils were dis-

tilled, and in each case a curve was plotted to temperature and per cent distilled as coordinates. These oils varied from very light engine oil to heavy aviation oil, some were used and some unused.

This course of investigation revealed, among other things, the fact that an oil sold as a 900 second viscosity oil was composed of fractions registering from below 100 seconds to over 4000 seconds, and that the residue was pitchy in nature. Curiously enough, this oil passed standard acceptance tests. Out of the entire lot of oils tested, only two showed anything approaching uniformity of composition.

These oils when distilled under a vacuum, sensibly a perfect one, showed absolutely no color change over a period of several months exposure to light and air. Identical oils distilled at atmospheric pressure were clear yellow when distilled, but blackened over night. This difference is attributed to decomposition at the higher temperature. What happens to this unstable material upon treatment with acid and alkali? It is not removed with the acid at all. Olefines form saturation products with the acid. Diolefines polymerize to a tar. Under the action of heat, oil will decompose first, which has been in its present molecular condition for ages or an unsaturated hydrocarbon compounded with sulphuric acid several months old. It is the opinion of the writer, based on results of the oils in use and under test, that the heavy sludge formed in use is due to the breaking down of the "purification products." This is supported by the positive data showing unusual consumption of oil every time a barrel is emptied into the system, containing approximately 450 gallons, indicating the presence of some portion of the oil which can not withstand the high temperatures of natural gas engines.

It is the belief of the writer that a vacuum distillation would be most effective in testing an oil for acceptance. It will show the presence of gasoline, kerosene, etc., on the light end, and heavy cylinder stocks and tar on the heavy end. Above all,

these are shown quantitatively and more satisfactorily than with a flash and fire test.

It is possible to mix undesirable light oils with undesirable heavy oils and obtain an oil which will pass acceptance tests. The answer to that subterfuge is a more rigid analysis, and vacuum distillation will effect it. The difference between a poor and a good oil is apparent by the curves shown.

The apparatus used in obtaining the data for these curves consisted of a 250 cubic centimeter distillation flask, a Liebig condenser, graduated receiver and a laboratory sized vacuum pump. The suction of the pump was connected to the graduated receiver, a vacuum of about one-tenth of a millimeter absolute pressure was pulled through the whole apparatus. The condensed vapors were collected in the receiver and all non-condensible vapors and gases were drawn out through the pump. A copper wire gauze in the form of a rolled sheet, extending from the bottom of the flask up to and beyond the delivery tube, was found indispensible to prevent "puking." Distillation was carried out otherwise in accordance with A.S.T.M. standards.

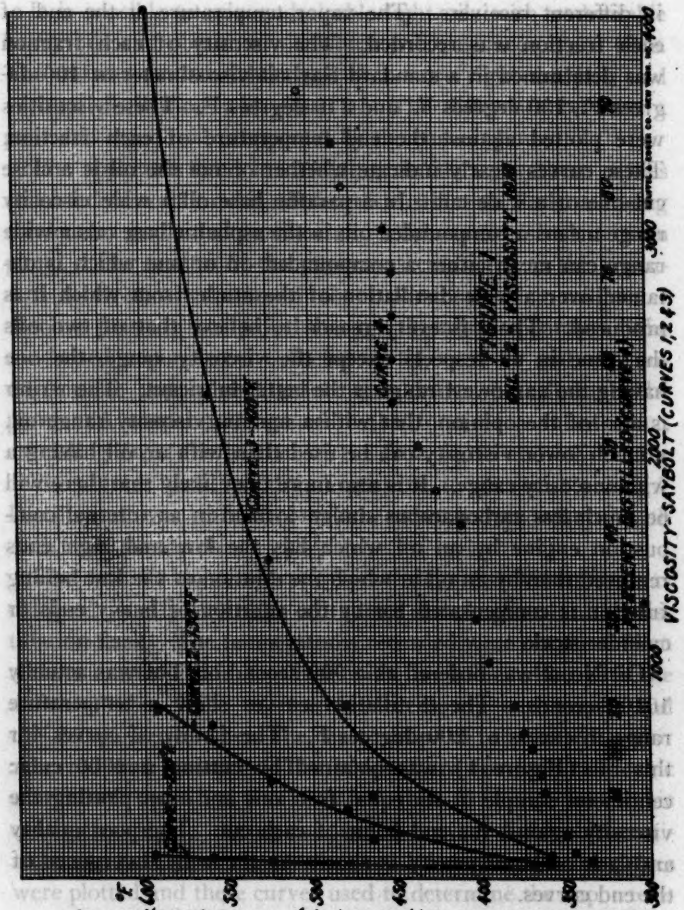
The data for the temperature-percentage distilled curves was obtained from a 100 cubic centimeter sample placed in the distillation flask. The temperatures recorded were obtained from a thermometer set in the flask with the bulb in line with the delivery tube. The oil was heated slowly and evenly and the temperature when the first drop of condensed vapor occurred was recorded. The temperature for each two cubic centimeters up to ten was recorded, then for every three cubic centimeters up to twenty-five and for every five cubic centimeters to the end of the distillation. From this data the curves were plotted and these curves used to determine the vapor temperature at which to make the fractionation cuts on larger samples.

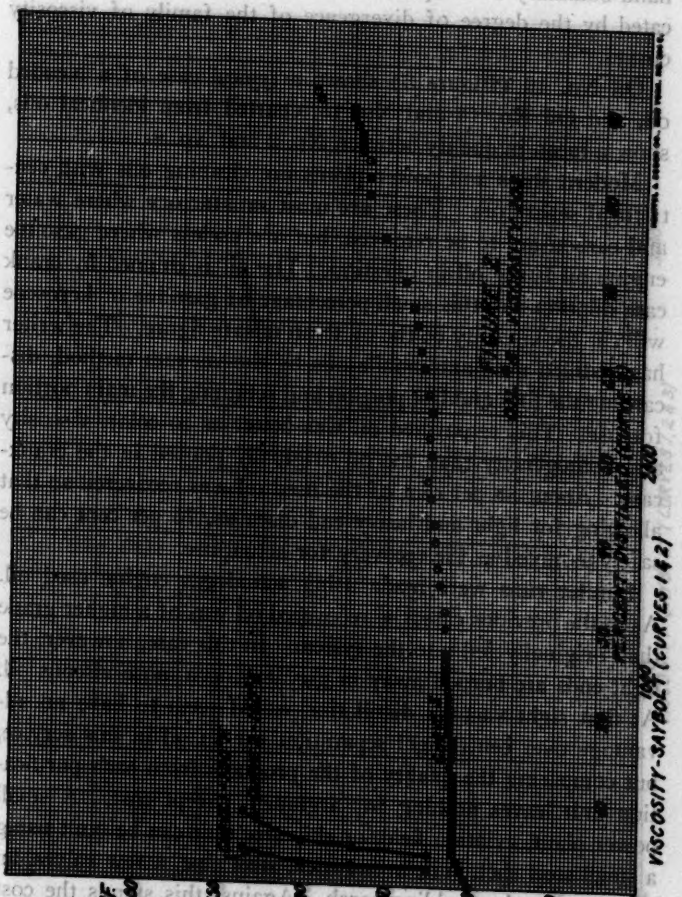
The fractionation of large samples was made in the same apparatus with the exception that a three thousand cubic centi-

meter distillation flask was substituted for the small one. A two liter sample was used and the various fractions collected in different receivers. The vapor temperature at the end of each fraction was recorded. The viscosity of each fraction was determined in a standard Saybolt viscosimeter at 100 degrees F., 130 degrees F., and 210 degrees F. These viscosities were plotted against the end temperature of each fraction. These curves clearly indicate whether or not the oil is a close cut one or a wide cut. In a paraffin base oil a wide viscosity range means a compounded oil, in the asphaltic base oils a wide range can mean either a compounded oil or one which is obtained over a wide distillation of the crude from which it is produced. There is every reason to believe that of two oils the same in all respects except the viscosity range, the one having the narrowest range is the better lubricant. The writer is also of the opinion, that with a narrow viscosity range oil, a much lower viscosity can be used than with an oil having a wide viscosity range. It is also more than likely that there will be much less carbonaceous sludge formed in an internal combustion engine by an oil which has the low and high ends removed than by an oil in which the dilution of the low boiling fraction is compensated for by the addition of heavy ends or cylinder stock.

Oil No. 2 was ordered as a 900 second oil, but was actually 1010 seconds. The distillation curves show a temperature range in excess of 200 degrees F. The family of curves for this oil (Figure 1) was obtained by cutting a 2000 cubic centimeter sample in six equal fractions and then plotting the viscosity against the end point of each cut. The poor quality and lack of homogeneity is shown by the wide divergence of the end curves.

Oil No. 8 shows an oil unusually perfect in character, very low in low ends and only a moderate amount of heavy ends (Figure 2). Note the approach to a family of vertical lines which would be the case in a perfect oil. It is to be appre-


VISCOSITY-SAYBOLT (CURVES 1,2 & 3)
END TEMPERATURE OF CUT, (CURVES 1, 2 & 3) - TEMPERATURE, (CURVE 4)


VISCOSITY - SAYBOLT (CURVES 1 & 2)
300
400
500
600
700
800
900
1000
END TEMPERATURE OF CUT (CURVES 1 & 2) - TEMPERATURE (CURVE 3)
650
600
550
500
450
400
350
300

ciated that a low viscosity oil will be charted close to the left hand boundary of the plot, but the character of the oil is indicated by the degree of divergence of the family of viscosity curves.

Oil No. 10 (Figure 3) shows a typical case of a blended oil, and Oil No. 14 and No. 15, charted from identical oils, show a mean in quality between No. 2 and No. 8.

Modern ships and power plants are cleaning oils with centrifugal separators. These are most satisfactory where water and carbon are to be removed, but are useless where gasoline engine oil is the oil in question. The oil is thinned by crank case dilution and this small percentage of gasoline or kerosene will cut the viscosity of the oil to an unsafe figure. The writer has distilled used automobile oils by the vacuum method, discarding the first and last portion and retaining the main portion for use. This re-purified oil was better as to color, viscosity and homogeneity than the oil originally poured in the crankcase. About 88 per cent of the used oil was recovered so that allowing for light and heavy end discards, 75 per cent can be safely assumed as the recovery for actual use.

Similar runs were made using actual gas engine used oil. As in the previous case, the recovered oil was of a higher grade than originally fed into the system. In this case, however, the light ends are burned away in use, leaving a more viscous oil. At first sight vacuum distillation would seem to have no advantage over centrifugal separation for an oil in this service, but comparing the quality of the products from each purification unit leaves nothing for argument except cost. A small power plant, aviation field, or ship size unit can be built using a small steel shell built in firebrick, steel pipe, a few valves, an air pump, and a welding torch. Against this stands the cost of a motor driven oil separator. The vacuum still would be operated on a gas or oil fired basis or by superheated steam if available. The separator uses electricity, so that operating cost favor the separator. But a vacuum still will reclaim aeroplane

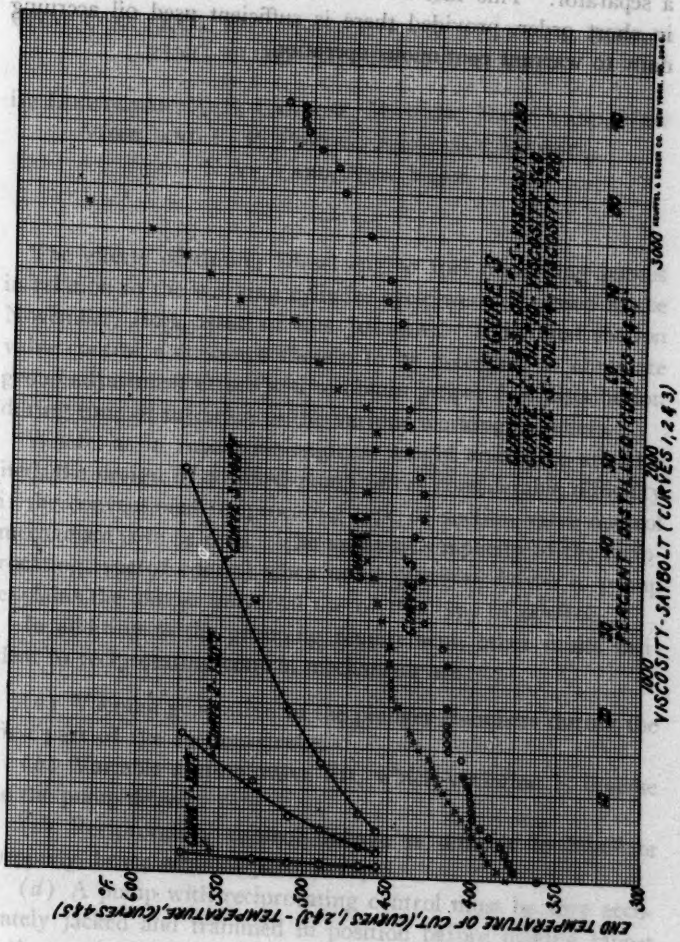
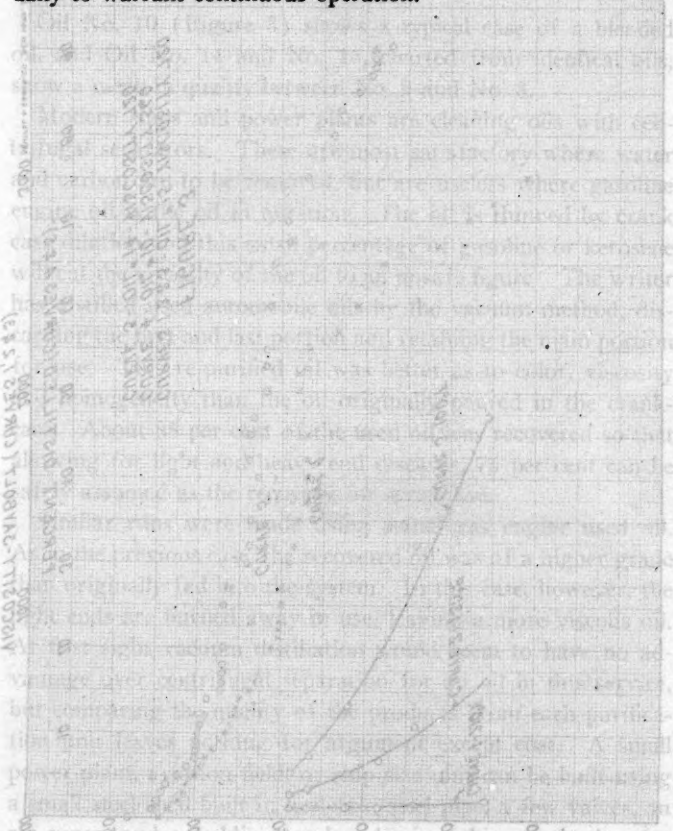


FIGURE 3
CURVE 1 - GAS OIL 0.850 GRAVITY 200
CURVE 2 - GAS OIL GRAVITY 0.860
CURVE 3 - OIL 0.870 VISCOSITY 200
PERCENT DISTILLED (CURVES 4 & 5)
VISCOSITY - SAYBOLT (CURVES 1, 2 & 3)
END TEMPERATURE OF CUT, (CURVES 1, 2 & 3) - TEMPERATURE, (CURVES 4 & 5)
Curve 1
Curve 2
Curve 3
Curve 4
Curve 5
600
550
500
450
400
350
300
200
1000
2000
3000
10
20
30
40
50
60
70
80
90

or motorboat engine oil which can not be made serviceable by a separator. This fact alone would pay for the vacuum still in short order, provided there is sufficient used oil accruing daily to warrant continuous operation.



CAM CONTROL OF DIESEL ENGINE FUEL MEASURING PUMP SUCTION VALVES.

By Lieutenant J. E. Canoose, Member, Navy Yard, New York, and F. P. Porter, Associate Member, Navy Yard, New York.

The effects of timing Diesel engine fuel measuring pumps in relation to the working cylinder cycle were discussed in the November, 1924, issue of this Journal. Individual suction valve control was shown therein to be necessary to eliminate group adjustment errors and to obtain proper fuel distribution during time of injection in the various working cylinders.

Where, as in submarine engines, weight and space are limited, the design of a timed pump offers considerable difficulty if the conventional reciprocating type of suction valve control mechanism is to be used. The apparent simplicity of the reciprocating type of control when using plungers driven in group explains the almost universal use of this type of control.

In addition to the grouping requirement, other faults incident to reciprocating control mechanism are:

(*a*) Vacuum generation in the plunger chambers during the first part of the suction stroke.

(*b*) Variable suction valve lift, which variation is inverse to the pump delivery rate.

(*c*) Reciprocating control linkage is difficult to lubricate properly, hence wear of parts is rapid.

(*d*) A pump with reciprocating control must be very accurately jacked and trammed in position before making tappet adjustments. This is very awkward and rarely satisfactorily done at sea. Experience shows that pumps are rarely trammed properly even under favorable conditions.

(*a*), (*b*) and (*d*) were discussed in previous articles* in this JOURNAL. (*c*) The difficulty involved in lubricating properly all the little points of friction in reciprocating control linkage is obvious.

Amongst possible forms of rotary control for pump suction valves, the camshaft seems to be most logical. This means offers suction valve control for any number of timed plungers with only one drive transmission from the pump crank shaft.

In addition to meeting the timing requirement in a practical manner, other factors must be considered:

(*a*) Vacuum generation within the fuel pump cylinders.

(*b*) Suction valve lift.

(*c*) Lubrication of control parts.

(*d*) Difficulties attending adjustment of tappet clearances.

(*e*) The effect of design on tappet clearance variation effects.

(*a*) Referring to previous discussion,† it is obvious that a valve control gear to provide against vacuum generation at all powers would have to open the suction valve at the beginning of the suction stroke at all powers. Were fifty per cent effective plunger travel to be used at full power, the control gear would have to hold the suction valve open 180 degrees of suction stroke crank travel plus 90 degrees of discharge stroke crank travel or a total of 270 degrees. But for closed throttle condition, it would be necessary to hold the suction valve open 360 degrees or throughout the total pump cycle. Obviously, to meet these requirements, a cam having a variable arc of lift would be required. The throttle in this case should simply change the arc of lift. This change of lift arc would have to affect only the point of closure of the valve if the point of opening of the suction valve were to be constant for all powers.

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* See pages 603 and 606 of the November, 1924, JOURNAL and page 37 of the February, 1925, JOURNAL.

† See page 601 of the November, 1924, JOURNAL.

In short, a cam capable of varying the point of valve closure at will without effecting the point of opening would provide an ideal suction valve control.

Unfortunately, no conventional type of cam possesses the above characteristics. It remains, therefore, to study the value of the conventional cam for use in the fuel measuring pump control gear, comparison being made with the performance of the reciprocating type of suction valve control.

In previous discussion,* it was pointed out that with reciprocating control the duration of vacuum generation in the pump plunger chamber during the first part of the suction stroke increases with the power developed. This allows less time for suction charging at full power than at low power. Due to the earlier closure of the suction valve at full power, there is less expulsion time for this condition than for lower power which lessens the possibility of completing suction charging during the expulsion angle of the discharge stroke. Reference to Figures 1, 2, and 3 makes this obvious.

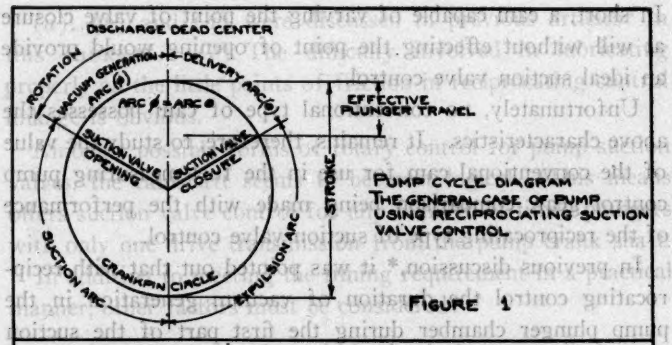
The fuel pump cycle diagram for the general case using reciprocating control of the suction valve is shown in Figure 1. Figures 2 and 3 are special cases of Figure 1, being that of closed throttle condition and that of full power condition respectively. The diagram in Figure 3 indicates the use of fifty per cent of discharge stroke for effective delivery. The effective plunger travel to be used on any design is dependent on several factors, a discussion of which will be found on page 62 of the February, 1925, JOURNAL.

From Figure 1 it is seen that *where reciprocating type of control is used the vacuum generation arc is unalterably equal to the delivery arc.* Both, therefore, increase as the amount of fuel being delivered increases. This statement remains true, regardless of the designed maximum effective stroke used.

In previous discussion† it was shown that very high velocities of fluid flow obtain in the *S-48* to *51* and *V-1* to *3* (2-cycle

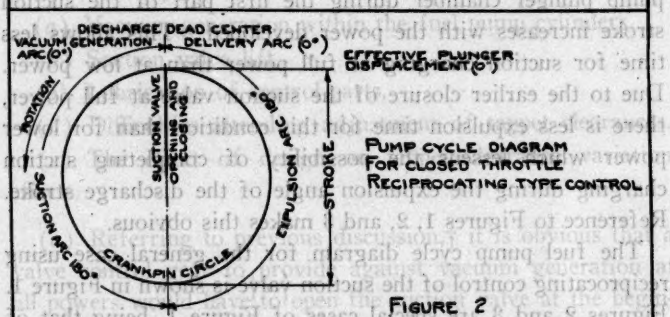
* See page 602 of November, 1924, JOURNAL.

† Pages 44 and 55 of February, 1925, JOURNAL.



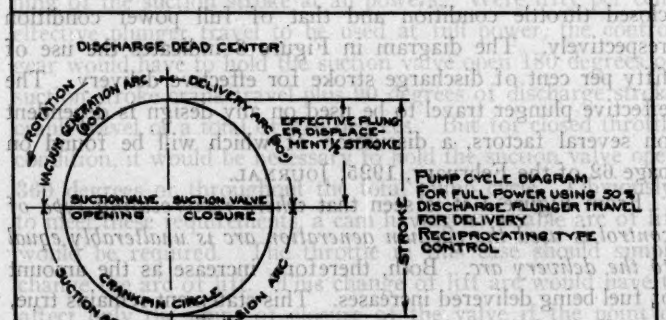
PUMP CYCLE DIAGRAM
THE GENERAL CASE OF PUMP USING RECIPROCATING SUCTION VALVE CONTROL

FIGURE 1



PUMP CYCLE DIAGRAM
FOR CLOSED THROTTLE
RECIPROCATING TYPE CONTROL

FIGURE 2



PUMP CYCLE DIAGRAM
FOR FULL POWER USING 50% DISCHARGE PLUNGER TRAVEL FOR DELIVERY
RECIPROCATING TYPE CONTROL

FIGURE 3

2500 H.P.) types of engine fuel pump suction valves. There is, therefore, little doubt that at full power, where only one-half of the suction stroke is available for charging, that the charging of fuel does continue during the expulsion arc. If charging does continue to the end of the expulsion arc, there is no actual expulsion and probably some degree of cavitation exists when suction valve closes. This partial cavitation and the failure to expel any air in the plunger chamber, admitted by leak or entrainment, is an important contributing factor to the very low volumetric efficiencies which are common to this type of pump in service.* The expulsion arc, especially at full power condition, should be available for expelling any air obtained by leaks or entrainment or gas liberated from the fuel.

Considering now the conventional cam for controlling fuel pump suction valves, it is obvious that it should be driven from the pump crankshaft at crankshaft speed. For the present a full power delivery arc of 90 degrees will be assumed. The cam function for full power obviously must be to open the suction valve at the beginning of the suction stroke and allow same to close at the beginning of the delivery arc, allowing valve to remain closed throughout the delivery arc. As 90 degrees of delivery arc corresponds to full power, for other powers it will be necessary to so move the closure point of the cam as to *reduce* the delivery angle. This will require the cam timing to be so changed as to allow the suction valve to close later in the pump cycle; but as the opening and closing points of the cam are fixed in relation to each other, it follows that retarding the point of closure will likewise retard the point of opening. Since at full power the suction valve is opened at the beginning of the suction stroke, any retardation of time of valve opening to reduce the delivery angle will result in creating a vacuum generation arc equal to such retardation arc. Hence, for 0 degrees delivery arc, 90 degrees of cam retardation will be required and 90 degrees of vacuum generation arc will obtain.

* Page 608 of November, 1924, JOURNAL.

Figures 4, 5 and 6 show the cam cycle for full power, any intermediate power, and closed throttle condition respectively. From these figures it is seen that at full power there is no vacuum generation arc which is the ideal condition. The vacuum generation arc increases as power is reduced until at closed throttle condition it equals the full power delivery arc. Obviously, the condition of no vacuum generation coincides with the minimum expulsion angle. With closed throttle condition when the vacuum generation arc is a maximum the expulsion arc is also a maximum which is a compensating characteristic incident to cam control.

Reference to Figures 1, 2, 3, 4, 5 and 6 makes the following tabulated characteristics for the limiting conditions apparent:

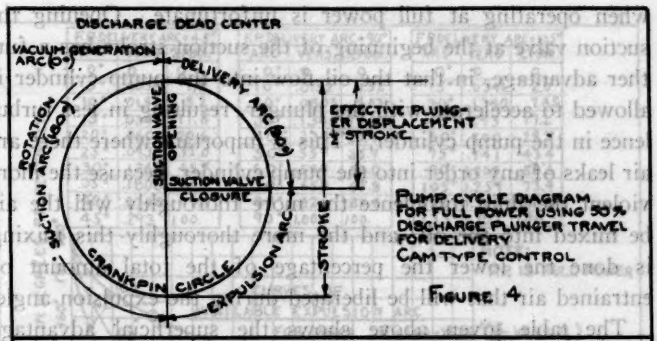
RECIPROCATING TYPE CONTROL.

	Time of Suction Valve Opening Degrees	Time of Suction Valve Closure Degrees	Open Suction Arc Degrees	Available Expulsion Arc Degrees	Vacuum Generation Arc Degrees
Full Power	90	270	180	90	90
Closed Throttle	0	360	360	180	0

CAM TYPE CONTROL.

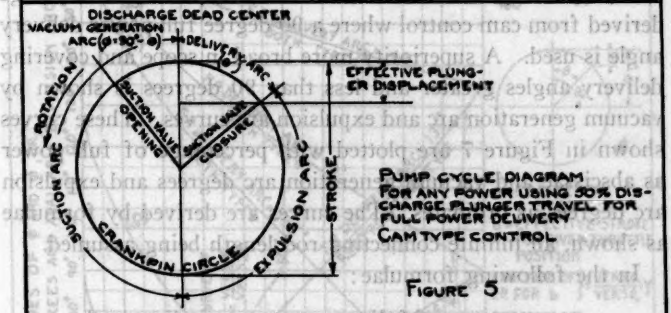
	Time of Suction Valve Opening Degrees	Time of Suction Valve Closure Degrees	Open Suction Arc Degrees	Available Expulsion Arc Degrees	Vacuum Generation Arc Degrees
Full Power	0	270	270	90	0
Closed Throttle	90	360	270	180	90

Obviously, the cam type control provides a constant total time for charging and expelling. The reduction of the total charging and expelling time by the reciprocating type control



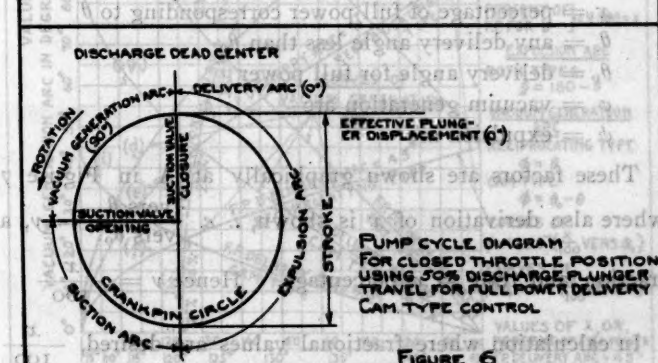
PUMP CYCLE DIAGRAM FOR FULL POWER USING 50% DISCHARGE PLUNGER TRAVEL FOR DELIVERY CAM TYPE CONTROL

FIGURE 4



PUMP CYCLE DIAGRAM FOR ANY POWER USING 50% DISCHARGE PLUNGER TRAVEL FOR FULL POWER DELIVERY CAM TYPE CONTROL

FIGURE 5



PUMP CYCLE DIAGRAM FOR CLOSED THROTTLE POSITION USING 50% DISCHARGE PLUNGER TRAVEL FOR FULL POWER DELIVERY CAM TYPE CONTROL

FIGURE 6

when operating at full power is unfortunate. Opening the suction valve at the beginning of the suction stroke has a further advantage, in that the oil flow into the pump cylinder is allowed to accelerate with the plunger, resulting in less turbulence in the pump cylinder. This is important where there are air leaks of any order into the pump cylinder, because the more violent the fluid turbulence the more thoroughly will the air be mixed into the oil; and the more thoroughly this mixing is done the lower the percentage of the total amount of entrained air that will be liberated during the expulsion angle.

The table given above shows the superficial advantage derived from cam control where a 90 degree full power delivery angle is used. A superiority more broad in scope and covering delivery angles greater and less than 90 degrees is shown by vacuum generation arc and expulsion arc curves. These curves shown in Figure 7 are plotted with percentage of full power as abscissae and vacuum generation arc degrees and expulsion arc degrees as ordinates. The curves are derived by formulae as shown, an infinite connecting rod length being assumed.

In the following formulae:

x = percentage of full power corresponding to θ
θ = any delivery angle less than θ_0
θ_0 = delivery angle for full power
φ = vacuum generation arc
ψ = expulsion arc

These factors are shown graphically at A in Figure 7 where also derivation of x is shown, *i. e.* $\frac{\text{vers } \theta}{\text{vers } \theta_0} = y$, a fraction, and $100y = x$, percentage. Hence $y = \frac{x}{100}$

In calculation where fractional values are desired, $\frac{x}{100}$ will be used instead of y, and thus the general equation will be

$$\frac{x}{100} = \frac{\text{vers } \theta}{\text{vers } \theta_0} \quad . \quad . \quad . \quad . \quad . \quad . \quad (1)$$

F.P. DELIVERY ARC = 45°		
ARC	VERS	%MAX.
0°	.0	0
5°	.0038	1.3
10°	.0152	5.2
15°	.0341	11.6
20°	.0603	20.6
25°	.0937	32.0
30°	.1340	45.7
35°	.1808	61.7
40°	.2340	79.9
45°	.293	100.

F.P. DELIVERY ARC = 90°		
ARC	VERS	%MAX.
0°	.0	0
10°	.0152	1.52
20°	.0603	4.03
30°	.1340	13.40
40°	.234	23.4
50°	.357	35.7
60°	.500	50.0
70°	.658	65.8
80°	.826	82.6
90°	1.000	100.

F.P. DELIVERY ARC = 135°		
ARC	VERS	%MAX.
0°	.0	0
15°	.0341	2.0
30°	.1340	7.85
45°	.293	17.2
60°	.500	29.3
75°	.741	43.4
90°	1.000	58.6
105°	1.259	73.9
120°	1.500	88.0
135°	1.707	100.

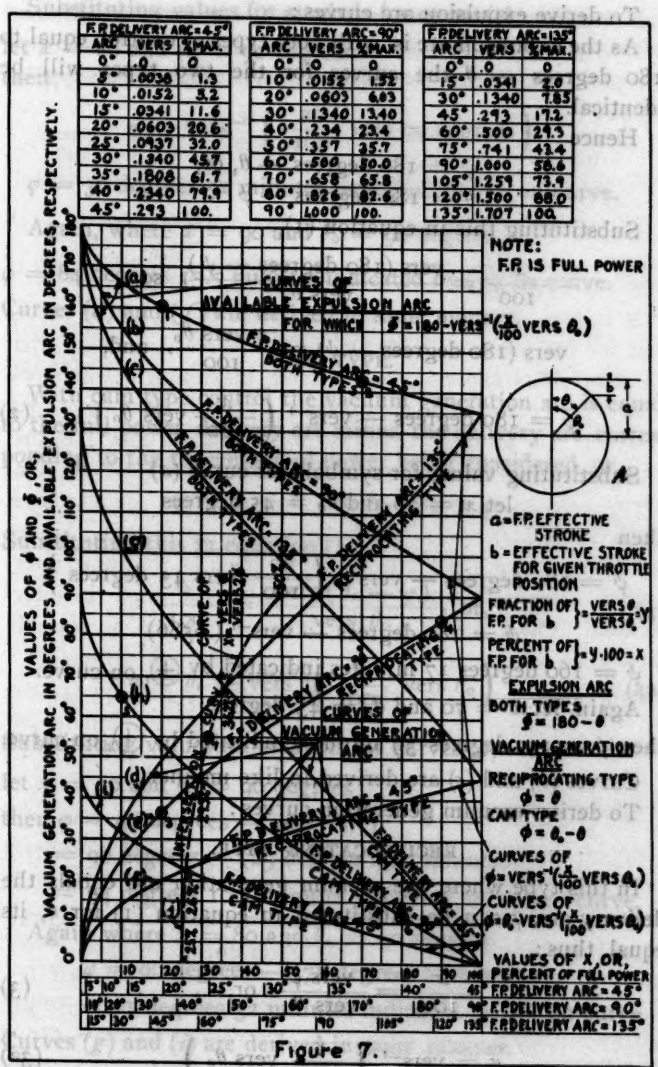

VALUES OF φ AND ψ, OR, VACUUM GENERATION ARC IN DEGREES AND AVAILABLE EXPULSION ARC IN DEGREES, RESPECTIVELY.
CURVES OF AVAILABLE EXPULSION ARC FOR WHICH ψ = 180 − VERS⁻¹(X/100 VERS θ)
F.P. DELIVERY ARC = 45° BOTH TYPES
F.P. DELIVERY ARC = 90° BOTH TYPES
F.P. DELIVERY ARC = 135° RECIPROCATING TYPE
F.P. DELIVERY ARC = 135° CAM TYPE
CURVE OF X = VERS θ
CURVES OF VACUUM GENERATION ARC
F.P. DELIVERY ARC = 45° RECIPROCATING TYPE
F.P. DELIVERY ARC = 45° CAM TYPE
F.P. DELIVERY ARC = 90° RECIPROCATING TYPE
F.P. DELIVERY ARC = 90° CAM TYPE
F.P. DELIVERY ARC = 135° CAM TYPE
NOTE: F.P. IS FULL POWER
a = F.P. EFFECTIVE STROKE
b = EFFECTIVE STROKE FOR GIVEN THROTTLE POSITION
FRACTION OF F.P. FOR b = VERS θ / VERS θ₁ = y
PERCENT OF F.P. FOR b = y·100 = X
EXPULSION ARC
BOTH TYPES ψ = 180 − θ
VACUUM GENERATION ARC
RECIPROCATING TYPE φ = θ
CAM TYPE φ = θ₁ − θ
CURVES OF φ = VERS⁻¹(X/100 VERS θ₁)
CURVES OF φ = θ₁ − VERS⁻¹(X/100 VERS θ₁)
VALUES OF X, OR, PERCENT OF FULL POWER
F.P. DELIVERY ARC = 45°
F.P. DELIVERY ARC = 90°
F.P. DELIVERY ARC = 135°

Figure 7.

To derive expulsion arc curves:

As the expulsion arc is, with both types of control, equal to 180 degrees — θ the curves for the two types will be identical.

Hence

$$\psi = 180 \text{ degrees} - \theta, \text{ or,}$$
$$\theta = 180 \text{ degrees} - \psi$$

Substituting this in equation (1)

$$\frac{x}{100} = \frac{\text{vers } (180 \text{ degrees} - \psi)}{\text{vers } \theta_0}, \text{ so,}$$

$$\text{vers } (180 \text{ degrees} - \psi) = \frac{x \text{ vers } \theta_0}{100}, \text{ and,}$$

$$\psi = 180 \text{ degrees} - \text{vers}^{-1} \left(\frac{x}{100} \text{ vers } \theta_0 \right) \quad . \quad (2)$$

Substituting values for symbols for curve (*a*)

let $x = 20$ and $\theta_0 = 45$ degrees

then

$$\psi = 180 \text{ degrees} - \text{vers}^{-1} \left(\frac{20}{100} \text{ vers } 45 \text{ degrees} \right)$$

$$\psi = 180 \text{ degrees} - \text{vers}^{-1} (.0586)$$

$\psi = 160$ degrees 17 minutes indicated by $\oplus_1$ on curve.

Again, let $x = 70$ and $\theta_0 = 45$ degrees

then $\psi = 142$ degrees 39 minutes indicated by $\oplus_2$ on curve.

Curves (*b*) and (*c*) are derived in like manner.

To derive vacuum generation curves:

RECIPROCATING TYPE.

In this type where the vacuum generation arc equals the delivery arc, φ can be substituted in equation (1) for θ, its equal, thus:

$$\frac{x}{100} = \frac{\text{vers } \varphi}{\text{vers } \theta_0}, \text{ or, } \quad . \quad . \quad . \quad (3)$$

$$\varphi = \text{vers}^{-1} \left(\frac{x}{100} \text{ vers } \theta_0 \right) \quad . \quad . \quad (3a)$$

Substituting values for symbols for curve (*e*)

let $x = 20$ and $\theta_0 = 90$ degrees.

then

$$\varphi = \text{vers}^{-1}\left(\frac{20}{100}\ \text{vers 90 degrees}\right)$$

$\varphi = 36$ degrees 52 minutes indicated by $\oplus_3$ on curve.

Again, where $x = 90$ and $\theta_0 = 90$ degrees

$\varphi = 84$ degrees 15.5 minutes indicated by $\oplus_4$ on curve.

Curves (*d*) and (*f*) are derived in same manner.

CAM TYPE.

With cam type control the vacuum generation arc is equal to the full power delivery arc minus the delivery arc corresponding to the percentage of power being considered, or,

$\varphi = \theta_0 - \theta$, and $\theta = \theta_0 - \varphi$

Substituting this in equation (1),

$$\frac{x}{100} = \frac{\text{vers}\,(\theta_0 - \varphi)}{\text{vers}\ \theta_0}, \text{ or, } \quad . \quad . \quad (4)$$

$$\varphi = \theta_0 - \text{vers}^{-1}\left(\frac{x}{100}\ \text{vers}\ \theta_0\right) \quad . \quad . \quad (4a)$$

Substituting values for symbols for curves (*h*)

let $x = 10$ and $\theta_0 = 90$ degrees

then $\varphi = 90$ degrees $-$ vers^{-1} (.1)

$= 90$ degrees $-$ (29 degrees 49.5 minutes)

$= 64$ degrees 10.5 minutes indicated by $\oplus_5$ on curve.

Again where $x = 80$ and $\theta_0 = 90$ degrees

$\varphi = 90$ degrees $-$ (78 degrees 28 minutes)

$=$ 11 degrees 32 minutes indicated by $\oplus_6$ on curve.

Curves (*g*) and (*i*) are derived in same manner.

Comparing now the vacuum generation curves (*d*), (*e*) and (*f*) with curves (*g*), (*h*) and (*i*), respectively, it is seen that the cam type control compares with the rceiprocating type as follows:

For Maximum Power Delivery Arc of: Degrees	Cam Gives Lower Vacuum Generation Arc from Full Power Down to: Per Cent
45	26
90	29.3
135	36.2

It is seen from the intersection of the two types of curves that for full power delivery arcs of 90 degrees and below, that the cam offers superior control from maximum power down to 29.3 per cent of maximum power. It should be remembered, however, that all these values refer to *indicated* H.P.; so that 29.3 per cent times the mechanical efficiency factor for that power becomes a very low figure. As has been stated above, delivery angles in excess of 90 degrees are rarely used. Values for delivery arcs other than the three given by the curve intersections may be found on curve (*j*) drawn through the intersections. Other points on this curve were found by formula derived as follows:

The basic formula for the reciprocating type curve is:

$$\frac{x}{100} = \frac{\text{vers } \varphi}{\text{vers } \theta_0}, \quad [(3) \text{ above}]$$

and for the cam type is

$$\frac{x}{100} = \frac{\text{vers } (\theta_0 - \varphi)}{\text{vers } \theta_0} \quad [(4) \text{ above}]$$

These curves will intersect when x and φ of one curve equal respectively x and φ of the other curve, or when

$$\frac{\text{vers } \varphi}{\text{vers } \theta_0} = \frac{\text{vers } (\theta_0 - \varphi)}{\text{vers } \theta_0}, \text{ or vers } \varphi = \text{vers } (\theta_0 - \varphi)$$

then $\theta_0 = 2\varphi$ . . . . . (5)

Hence for any value of θ_0 the intersection of the two curves will fall on the curve found by substituting in the reciprocating basic equation thus:

$$\frac{x}{100} = \frac{\text{vers } \varphi}{\text{vers } 2\varphi} \text{ or } x = 100 \frac{\text{vers } \varphi}{\text{vers } 2\varphi}$$

Substituting in this for intersection where

$\theta_0 = 90$ degrees, from (5) $\varphi = 45$ degrees, so we find

$$x = 100 \frac{\text{vers } 45 \text{ degrees}}{\text{vers } 90 \text{ degrees}} = 100 \frac{(.29299)}{1} = 29.299 \text{ per cent}$$

Again, suppose $\theta_0 = 110$ degrees, then $\varphi = 55$ degrees

Substituting

$$x = 100 \frac{\text{vers } 55}{\text{vers } 110} = 100 \frac{.42642}{1.34202} = 31.77$$ indicated by $\oplus_7$ on curve (j).

Substitution may likewise be made in the basic formula for the cam type, for as $\theta_0 = 2\varphi$, then

$$\frac{x}{100} = \frac{\text{vers } (2\varphi - \varphi)}{\text{vers } 2\varphi} = \frac{\text{vers } \varphi}{\text{vers } 2\varphi}$$

Below the line of abscissa values for the curves titled "Per cent of full power" are shown three other lines giving to scale the angles corresponding to the power percentages for the delivery arc angles used in the curves.

Summarizing, the following points have been discussed. The cam type control gives an ideal condition for vacuum generation and expulsion at full power varying from the ideal at lower powers by increased vacuum generation which, however, is compensated for by increased expulsion. The reciprocating type control gives the ideal condition at closed throttle varying from the ideal by increased vacuum generation at higher powers. The maximum vacuum generation occurs at full power, at which time the expulsion is the least. Furthermore, the vacuum generation for the cam type control for any full power

effective stroke is less than for the reciprocating type control at full power and also over nearly the entire power range. Summarily, therefore, the cam control gives a markedly superior vacuum generation factor.

(*b*) The variable suction valve lift incident to reciprocating control has been discussed.* Obviously, the cam provides an unvarying lift.

(*c*) It is fairly simple to provide an oil bath for lubricating all parts of a cam shaft. In fact it is not difficult to provide positive lubrication for all working parts in the fuel measuring pump where cam control is used. The lubrication of working parts in reciprocating control linkage is impracticable and in practice is almost always inadequate.

(*d*) The difficulties attending adjustment of tappet clearances in reciprocating types of control have been discussed at some length.† Obviously the variable lift of suction valve incident to reciprocating control results in variable tappet clearance. The engine must therefore be jacked to some predetermined point for adjustment of each plunger unit. It is equally obvious that with a cam controlled fuel pump it would only be necessary to jack the engine to any point where the roller is off the cam. This can be further simplified by throttle manipulation of the cam timing.

(*e*) Assuming a pump is designed which has all the fundamental errors of design removed, such as: vacuum generation; high fluid flow velocities through suction (or discharge) valves; grouped plungers; plunger packing failures; valve tossing; high cyclic speeds; adjustment difficulties; and other factors reducing the volumetric efficiency of the pump; there remains a factor over which the operator has little control and which at times takes on proportions astonishing to even the best of operators. This is the effect of tappet clearance variation on the delivery balance between the various pump units.

* Page 606 of November, 1924, JOURNAL.

† Page 603 of November, 1924, JOURNAL.

If the above listed factors are properly met, the pump would very nearly fulfill the ideal of delivering exactly the same quantity of fuel to each working cylinder, thereby evenly distributing the load between them, *only so long as the suction valve tappet absolute clearances were kept perfectly uniform.* Absolute clearance error is taken to mean the clearance remaining after such defects as machining errors, deflections of elastic parts and effects due to inertia have been removed—in other words, actual operative clearances.

Obviously the machining error defects must be removed by the manufacturing machinist, and the designer must provide such a design that sufficient strength will reduce elastic deflections in pump frame, shafts, and suction valve control parts to a low value, and he must further provide such symmetry of design as to equalize as nearly as possible the residual elasticity of such parts. This is assumed done.

Now there remains the question of uniformity of adjustment and *wear* of tappets after adjustment. In previous discussion (pages 602-606 of November, 1924, JOURNAL) it was shown that uniform group tappet adjustments were not obtained even on the test block where conditions are ideal. There is further little doubt that some of the power unbalance, discussed on pages 610-611 of above mentioned JOURNAL, was due to non-uniform tappet clearances. In this discussion of engine number 581 (*S-48* to *51* class) it was shown that when running on the test block, at overload power, cylinder number 5 carried 140 pounds M.E.P. while cylinder number 3 carried 70 pounds M.E.P. In view of such block test performances, one should hardly expect the constantly shifting Navy operating personnel to make adjustments of tappets uniform; but *assuming* that at the beginning of every cruise every submarine engine has uniformly adjusted tappets, they will not long stay uniform, due to wear of the tappets themselves and due in greater degree to wear of the suction valves into their seats. If cruises are short, adjustments are made often enough to

keep the engines running fairly well; but when several days of continuous running is required, after a few days the engine begins to sound "lame." Inspection at these times invariably shows that tappets need readjustments which is usually done in such manner that the smoothness of engine operation after adjustment is directly proportional to the smoothness of the sea when the adjustment was made.

Now this may be all true but what shall be done about it other than to minimize the wear? The designer before selecting a suction valve control should study the effect of possible differences in tappet clearances on the uniformity of fuel delivery with a view to selecting a design which will reduce clearance variation effects to a minimum.

Clearance differences resulting from wear very commonly are found to be .004 inch, and in many cases exceed that even with best designs and operators. Investigation of the designer should therefore cover clearance differences of at least .004 inch.

In previous discussion (February, 1925, JOURNAL, pages 59 to 70 incl.) the effect of design on tappet clearance variation effects was shown for reciprocating types of control. The quantity of fuel delivered is supposed to be equal to the plunger displacement after the instant of suction valve closure.

Figure 8, a fuel measuring pump cylinder indicator card, is here reproduced from Figure 6 of the February, 1925, JOURNAL, page 48. As this card shows at (*d*), the pressure in the pump cylinder is not built up instantaneously. Due to wire-drawing, the pressure starts to build up as soon as the suction valve nears its seat. The rate of building this pressure is dependent on the ratio of plunger displacements to suction valve area, *i.e.*, the larger the suction valve area in proportion to the plunger displacement rate, the less the building of pressure that will occur due to wire-drawing the fuel in passing the suction. Obviously, therefore, the pressure building rate or the pressure rise fillet shown on the indicator card will have different values in any one design for different delivery angles.

It would decrease in value with decreased delivery angles in proportion to the sine curve of plunger velocity if the rate of valve closure were constant and therefore independent of plunger velocity. But in reciprocating types of control the rate of valve closure is dependent on the plunger velocity being directly proportional thereto because of receiving its drive from the plunger driver. Therefore, the indicator card pressure rise fillet would decrease in value with decreased values of delivery angles in proportion to the resultant of the changing valve closure rate factor and the plunger velocity factor.

With cam control, the valve closure rate will remain constant for all delivery angles. With this type, therefore, the indicator card pressure rise fillet will decrease with decreased delivery angle values in proportion to the sine curve of plunger velocity. If, therefore, the cam be so designed that the closure velocity rate is equal to the maximum closure velocity rate for the reciprocating control, the card fillet values for the cam control would tend to run lower than reciprocating control fillet values as delivery angles were decreased; and as the delivery angle approached zero the values for both would approach zero because for both types the fillet value should be zero at zero delivery angle. Where 90 degrees full power delivery angles are used, the delivery angle for idling condition is seldom below 30 degrees; so indicator card fillet characteristics below this point need no consideration.

Before discussing the value to the designer of this pressure rise filleting as shown by the indicator card a dimensional description of the fillet characteristics will be given. Figure 9 is a large scale reproduction of the average fillet found in cards taken on the Bureau type 1000 B.H.P. engine operating at such powers that a delivery angle of 90 degrees was used. Since at 90 degrees crank position the plunger travel equals the peripheral speed of the pump crank pin, the fillet abscissa value can be converted directly to crank travel. This was done as shown, as is also the corresponding valve travel, obliquity

of the connecting rod being neglected in each case. As this figure shows, a pressure of 36 pounds was built up in the pump cylinder before the valve completely closed. The suction valve effective pressure area* in this case was about .234 square inches, hence, $.234 \times 36 = 8.43$ pounds reactive pressure against valve control gear. In other words, at point of valve closure there is 8.4 pounds hydrostatic pressure augmenting the suction valve spring pressure. This 8.4 pounds is, of course, a maximum which scales down the fillet curve to 0, where fillet begins.

For reciprocating control, this pressure filleting before valve closure has no use. In fact it sets up additional stress in the valve control gear, which requires greater strength and symmetry of control gear parts. For cam control, the problem is different, and this pressure filleting can be used to give unique possibilities for cam control performance. This will be considered in the following discussion of the cam control.

Cam control of any part of a mechanism is limited by the forces available for making the cam follower keep contact with the cam profile. Inertia of parts and allowable spring pressures are the limiting factors in designing the cam profile. Where spring pressure is constant, a uniform acceleration cam profile curve is desirable for taking care of inertia.

The cam profile chart shown in Figure 11 shows the increases in cam radii, on a straight line base, over the closure arc of the cam, assuming the cam roller to be a point. The abscissa in this chart represents degrees of cam rotation. The cam roller, however, is not ordinarily assumed a point. In most designs it is comparatively large. The larger the roller diameter in proportion to the came size, the greater will be the difference in contour between cam profile curve and the pitch surface curve, which is the line connecting successive roller pin center positions imagining the line to be scribed on a plane rotating with the cam over any given arc of cam rotation. The chart

* See page 38 of February, 1925, JOURNAL.

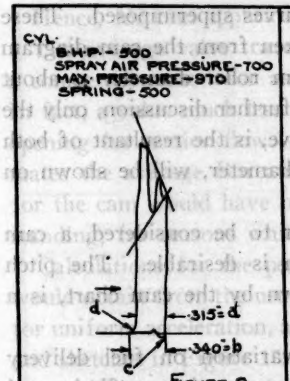


FIGURE 8

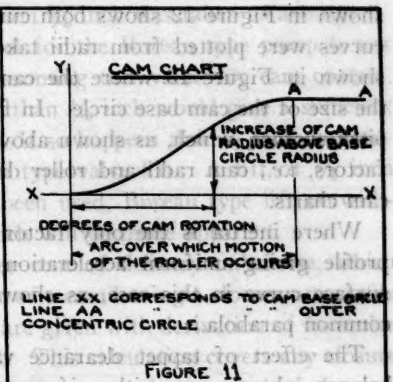
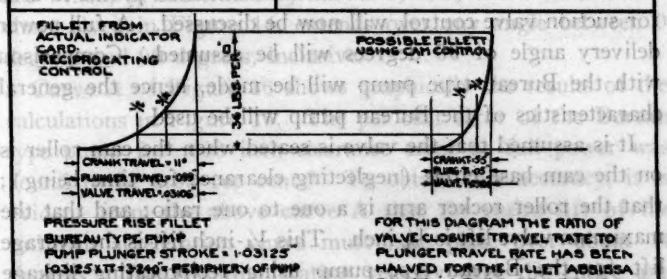


FIGURE 11


FILLET FROM ACTUAL INDICATOR CARD. RECIPROCATING CONTROL
POSSIBLE FILLET USING CAM CONTROL
CRANK TRAVEL = 11°
PLUNGER TRAVEL = .099"
VALVE TRAVEL = .03106"

PRESSURE RISE FILLET
BUREAU TYPE PUMP
PUMP PLUNGER STROKE = 1.03125"
1.03125 x π ÷ 3.240" = PERIPHERY OF PUMP CRANK PIN CIRCLE.
$\frac{3.240}{360}$ = .0090" INCHES PER DEGREE OF PERIPHERY
.0090 x 11 = .099" = PLUNGER TRAVEL CORRESPONDING TO FILLET ABSCISSA.
1.31875 = RATIO OF VALVE TRAVEL TO PLUNGER TRAVEL.
$\frac{.099}{3.1875}$ = .03106 = VALVE TRAVEL CORRESPONDING TO FILLET ABSCISSA, AS THE DISTANCE VALVE IS FROM ITS SEAT WHEN FILLETING COMMENCES.

FIGURE 9

FOR THIS DIAGRAM THE RATIO OF VALVE CLOSURE TRAVEL RATE TO PLUNGER TRAVEL RATE HAS BEEN HALVED. FOR THE FILLET ABSISSA RANGE, i.e. THE CLOSURE TIME OVER THE FILLET RANGE HAS BEEN HALVED. HERE THE CORRESPONDING POINTS IN ABSCISSA (½, ¾ ETC.) HAVE EQUAL ORDINATE VALUES SINCE THE CHARACTERISTIC OF PRESSURE DROP ACROSS VALVE FOR ANY GIVEN OPENING AND EXPULSION VELOCITY REMAINS UNCHANGED.

FIGURE 10

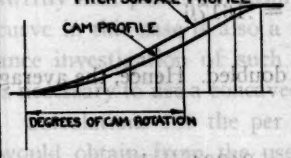


FIGURE 12

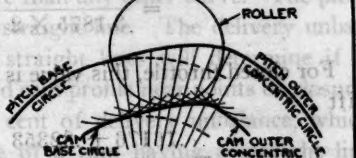


FIGURE 13

shown in Figure 12 shows both curves superimposed. These curves were plotted from radii taken from the cam diagram shown in Figure 13 where the cam roller diameter is about the size of the cam base circle. In further discussion, only the pitch surface, which, as shown above, is the resultant of both factors, *i.e.*, cam radii and roller diameter, will be shown on cam charts.

Where inertia is the only factor to be considered, a cam profile giving uniform acceleration is desirable. The pitch surface curve in this case, as shown by the cam chart, is a common parabola.

The effect of tappet clearance variation on fuel delivery balance, where a cam with uniform acceleration profile is used for suction valve control, will now be discussed. A full power delivery angle of 90 degrees will be assumed. Comparison with the Bureau type pump will be made, hence the general characteristics of the Bureau pump will be used.

It is assumed that the valve is seated when the cam roller is on the cam base circle (neglecting clearance for time being); that the roller rocker arm is a one to one ratio; and that the maximum valve lift is ¼ inch. This ¼-inch lift is the average lift for the Bureau type pump using reciprocating linkage and is derived as follows:

Lift of valve for full power

$$= \frac{\text{valve travel to plunger travel ratio} \times \text{stroke}}{2}$$

substituting*

$$= \frac{1 \times 1.03125}{3.1875 \times 2} = .16176$$

For closed throttle, this value is doubled. Hence, the average lift

$$= \frac{.16176 + .32353}{2} = .24264 \text{ inch}$$

* See page 60 of February, 1925, JOURNAL.

Hence, for cam application ¼ inch was taken as average.

The average valve travel for the Bureau type pump is not a competent reason for assuming this value for cam control. In fact, a value much less than ¼ inch would give a sufficient opening for suction flow. It so happens that ¼ inch approximates the average Bureau type valve lift. A lift of ¼ inch for the cam would have been used, Bureau type lift notwithstanding, for reasons which will be made apparent.

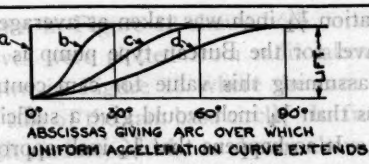
Calculations for the per cent of delivery unbalance, which would obtain from the use of a cam having a closure arc profile for uniform acceleration, are given with derivation of formulae and sketches in Figure 15. This calculation covers any closure arc which, as illustrated by curves *a, b, c,* and *d* of the cam chart given in Figure 14, extends over any angle between 0 degrees and 90 degrees, inclusive.

Curve A of Figure 16 shows graphically the results of the calculations in Figure 15. The per cent error of the Bureau type pump for .003 inch clearance error at full power was found to be 1.85 per cent.* To obtain this accuracy, it is evident from the above curve A that the entire lift of the uniform acceleration curve must take place over about 14 degrees. The resulting seating velocity over the entire arc, however, is much too swift, except for the point of closure where there is a large force available to hold the cam and cam follower in contact. It is therefore concluded that a uniform acceleration curve used in this way is undesirable.

Of the possible cam profiles without re-entrant angles, the straight line tangent to the inner base circle falls off more swiftly near points of closure than any other curve. The pitch curve in this case is also a straight line. The delivery unbalance investigation of such straight lines will determine if it is necessary to use a concaved cam profile near points of closure.

Calculations for the per cent of delivery unbalance, which would obtain from the use of a cam having a straight line

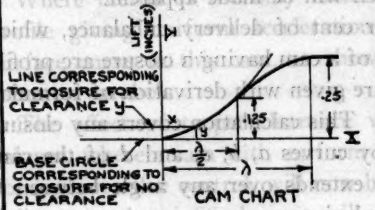
* See page 60 of February, 1925, JOURNAL.



CURVE	ARC OVER WHICH CURVE EXTENDS
a	0°
b	30°
c	60°
d	90°

FIGURE 14

CALCULATIONS FOR PERCENT ERROR OF UNIFORM ACCELERATION CURVES



CAM CHART

X = DEGREE OF CAM ROTATION

Y = CORRESPONDING LIFT OF CAM

x = CLEARANCE ERROR ANGLE CORRESPONDING TO y

y = CLEARANCE ERROR

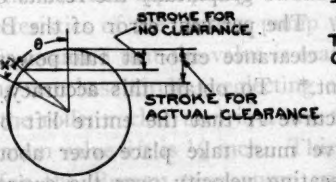
λ = ARC OVER WHICH UNIFORM ACCELELERATION CURVE EXTENDS

THE EQUATION OF THE PARABOLA FROM 0 TO $\frac{\lambda}{2}$ IS

$$X^2 = \frac{\left(\frac{\lambda}{2}\right)^2}{\cdot 125} Y, \text{ OR, } X^2 = 2\lambda^2 Y$$

THEREFORE $X^2 = 2\lambda^2 y$

OR $X = \lambda\sqrt{2y}$



CRANK DIAGRAM SHOWING EFFECT OF CLEARANCE ERROR ANGLE (x) UPON DELIVERY

θ = DELIVERY ANGLE FOR NO CLEARANCE

PERCENT ERROR IN DELIVERY DUE TO DIFFERENCE IN CLEARANCE AT TAPPET FOR CAM CONTROLED SUCTION VALVE =

$$100\left[\frac{\text{VERS}(\theta+x)}{\text{VERS}\,\theta} - 1\right]$$

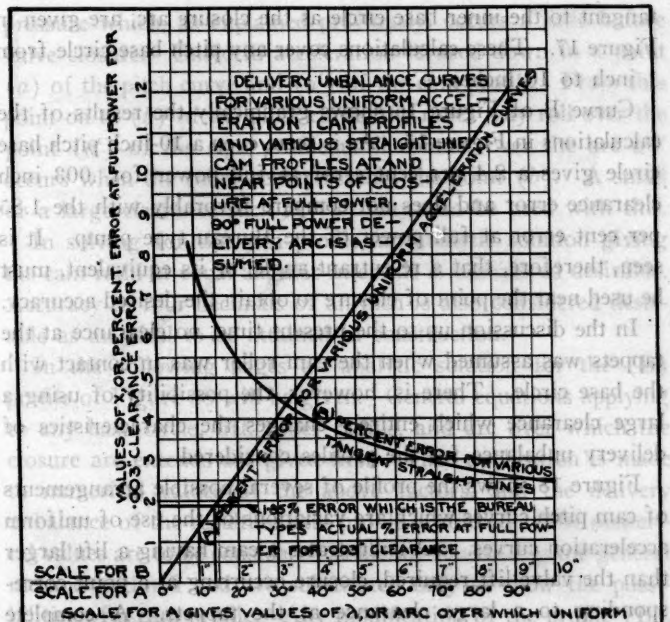
PERCENT ERROR FOR UNIFORM ACC. CURVES FOR CLEARANCE ERROR OF ·003"

λ	x FOR y = ·003	VERS $(\theta+x)$	PERCENT ERROR
90°	6·9′	1·121	12·1
75°	5·61°	1·101	10·1
60°	4·64°	1·081	8·1
45°	3·48°	1·061	6·1
30°	2·32°	1·040	4·0
15°	1·16°	1·020	2·0
0°		1.	0

PERCENT ERROR PLOTTED ON λ IS FOUND IN FIGURE 16 CURVE A

FIGURE 15

* See page 60 of February, 1925, Journal.


DELIVERY UNBALANCE CURVES FOR VARIOUS UNIFORM ACCELERATION CAM PROFILES AND VARIOUS STRAIGHT LINE CAM PROFILES AT AND NEAR POINTS OF CLOSURE AT FULL POWER. 90° FULL POWER DELIVERY ARC IS ASSUMED.
VALUES OF X, OR PERCENT ERROR AT FULL POWER FOR .003" CLEARANCE ERROR.
PERCENT ERROR FOR VARIOUS UNIFORM ACCELERATION CURVES
PERCENT ERROR FOR VARIOUS TANGENT STRAIGHT LINES
1.85% ERROR WHICH IS THE BUREAU TYPES ACTUAL % ERROR AT FULL POWER FOR .003" CLEARANCE.
SCALE FOR B
SCALE FOR A
0° 10° 20° 30° 40° 50° 60° 70° 80° 90°
1 2 3 4 5 6 7 8 9 10
1 2 3 4 5 6 7 8 9 10 11 12

SCALE FOR A GIVES VALUES OF λ, OR ARC OVER WHICH UNIFORM ACCELERATION CURVE EXTENDS.

SCALE FOR B GIVES VALUES OF η OR RADIUS OF PITCH BASE CIRCLE

FIGURE 16

CALCULATIONS FOR PERCENT ERROR FOR TANGENT STRAIGHT LINE.

X = DEGREES OF CAM TRAVEL TO TAKE UP CLEARANCE

r = RADIUS OF PITCH BASE CIRCLE

CLEARANCE = .003"

$$\sec X = \frac{r + .003}{r} = 1 + \frac{.003}{r}$$

PERCENT ERROR FOR VARIOUS VALUES OF r

r"	SEC X	TAN X	X	VERS (90-X)	PERCENT ERROR
1"	1·0030	·006	4°26'	1·077	7·7
2"	1·0015	·003	3°8'	1·055	5·5
3"	1·0010	·002	2°34'	1·045	4·5
4"	1·00075	·0015	2°13'	1·039	3·9
5"	1·0006	·0012	1°59'	1·035	3·5
6"	1·0005	·0010	1°49'	1·032	3·2
10"	1·0003	·0006	1°24'	1·024	2·4

PERCENT ERROR PLOTTED ON r IS CURVE B IN THE ABOVE CHART

FIGURE 17

tangent to the inner base circle as the closure arc, are given in Figure 17. These calculations cover any pitch base circle from 1 inch to 10 inches.

Curve B of Figure 16 shows graphically the results of the calculations in Figure 15. As is seen, even a 10-inch pitch base circle gives a 2.4 per cent error at full power for .003 inch clearance error and does not compare favorably with the 1.85 per cent error at full power of the Bureau type pump. It is seen, therefore, that a re-entrant angle, or its equivalent, must be used near the point of closure to obtain the desired accuracy.

In the discussion up to the present time, no clearance at the tappets was assumed when the cam roller was in contact with the base circle. There is, however, the possibility of using a large clearance which entirely changes the characteristics of delivery unbalance for the profiles considered.

Figure 18 shows the profile of several possible arrangements of cam pitch curves which are variations of the use of uniform acceleration curves. (A) illustrates a cam having a lift larger than the valve lift required, closure occurring at a point corresponding to a large clearance at the tappets. A complete uniform acceleration curve is used. (B) illustrates a cam having a lift equal to the valve lift. A portion of a uniform acceleration curve is used over the arc which the valve closure extends, the base circle continuing the cam profile at point of valve closure. No clearance is anticipated in this diagram, which, of course, is merely the limiting case of a small clearance. No difficulty should be encountered with judicial use of this type of profile, for, as explained previously, there is an augmentive force at point of closure holding the valve seated, so that contact of the roller with the cam should result. (C) is a modification of (B) greatly exaggerated, a fillet being used at (*a*) between the uniform acceleration pitch curve and the base circle. This may also be used with a clearance such as (*b*). (D) is still another modification of (B). This profile may be designed to take advantage of the augmented spring

pressure which, as explained previously, occurs a little before valve closure. Uniform acceleration is used down to a point (a) of the pitch curve and (a') of the cam profile. From this point on to (b) of the pitch curve, the roller may roll over the point (a') of the cam profile until contact over the arc $a'b'$ occurs when the roller center reaches the point (b). A small or a large clearance such as (d) may also be used with this.

In solving for the cam profile, an analytic solution giving the cam radii at equal angular intervals is considered desirable. Accuracy to four decimals of an inch is also considered desirable as an ideal to be attained in construction.

In the following pages an analytic solution for the (B) profile of Figure 18 will be given. General equations applying to any base circle, lift, roller radius, and arc over which the closure arc extends are given first. Then application is made to numerical values of the above factors, and the delivery unbalance of the resulting cam profile is studied. The numerical values are to be considered as merely illustrating the general method outlined and are substituted only to show the possibilities of a cam profile. Any modification of the profile (B), such as shown at (A), (C) or (D) of Figure 18, may be easily obtained by a slight change in the method outlined.

Let ρ_1 and α_1 be the polar co-ordinates of any point on the pitch curve (refer to Figure 19) measuring the angle α_1 from any convenient radius, and let ρ and α be the polar co-ordinates of a point on the cam profile, which is the point of tangency of the roller when its center is on the point (ρ_1, α_1). Let α be measured from the same reference line as α_1. Consider the pitch curve known and to have the form

$$\rho_1 = f(\alpha_1)$$

To obtain the relation between ρ, α, ρ_1 and α_1, consider the triangle CDO. Letting r = the roller radius, we have

$$\rho^2 = \rho_1^2 + r^2 - 2\,\rho_1\, r \cos \angle \text{DCO} \quad . \quad . \quad (6)$$

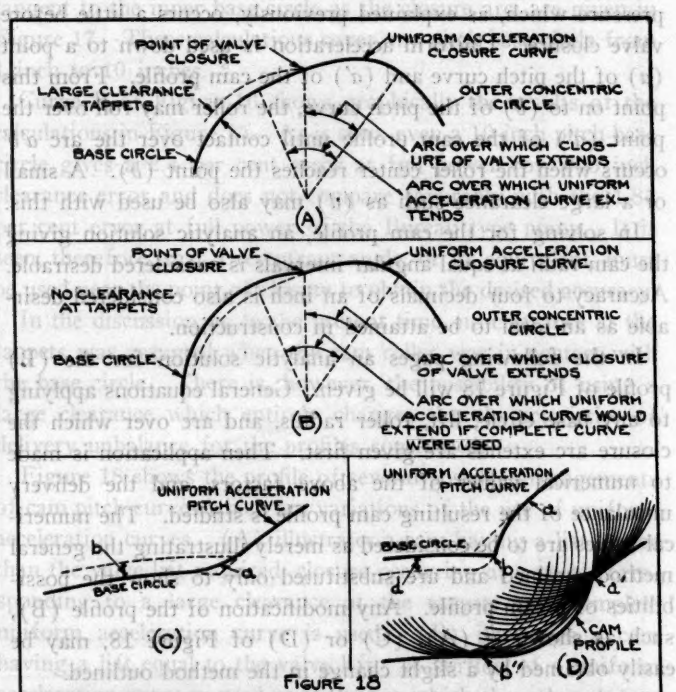


FIGURE 18

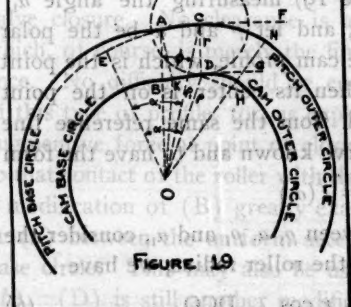


FIGURE 19

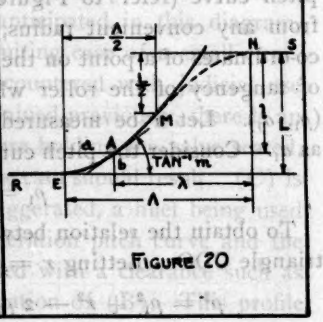


FIGURE 20

Now, $\angle$ DCO is the angle between the radius vector and the normal to the pitch curve at any point. Hence,

$$\tan \angle \text{DCO} = \frac{d\rho_1}{da_1} \cdot \frac{1}{\rho_1}$$

and the cosine may be written

$$\cos \angle \text{DCO} = \frac{1}{\sqrt{\left(\frac{d\rho_1}{da_1} \cdot \frac{1}{\rho_1}\right)^2 + 1}} = \frac{\rho_1}{\sqrt{\left(\frac{d\rho_1}{da_1}\right)^2 + \rho_1^2}}$$

Therefore, substituting in (6) the analytic solution of radial distances on the cam profile corresponding to any given pitch curve consists in solving the equations

$$\rho^2 = \rho_1^2 + r^2 - \sqrt{\frac{4\rho_1^4 r^2}{\left(\frac{d\rho_1}{da_1}\right)^2 + \rho_1^2}} \quad . \quad . \quad . \quad (7)$$

where

$$\rho_1 = f(a_1) \quad . \quad . \quad . \quad . \quad (8)$$

and

$$\cos(a - a_1) = \frac{\rho^2 + \rho_1^2 - r^2}{2\rho\rho_1} \quad . \quad . \quad . \quad (9)$$

Equation (8) gives ρ_1 for any value of a_1, equation (7) gives ρ for any values of ρ_1 and a_1, and equation (9) gives the corresponding value of a.

The function f in equation (8) for a uniform acceleration curve is obtained as follows:

Two equal parabolas make up the pitch surface in the cam chart such that they are tangent at their vertices with lines corresponding to inner and outer concentric circles, and also are tangent to each other at one-half the lift. Refer to the cam chart given in Figure 20. Lines ER and NS correspond to inner and outer concentric circles respectively of the cam pitch surface. Curve EM is a parabola tangent at its vertex E to ER and tangent at M to an equal parabola MN, which is in turn tangent to NS at its vertex N. Assuming the cam

exists in a developed form below the inner concentric circle ER, the parabolas, EM and MN, extend from their vertices away from and toward the cam respectively.

Let $\wedge$ = Arc over which the complete uniform acceleration curve extends
L = Lift of complete uniform acceleration curve
x = Angle co-ordinate of the pitch curve
y = Lift co-ordinate of the pitch curve
P = Constant in the equation of the parabola
R = Radius in the actual cam pitch curve of the inner or outer concentric circles.

Let the vertices of the parabolas be the origin for x and y for each parabola. Therefore, the equation of the parabolas is

$$x^2 = Py \quad . \quad . \quad . \quad . \quad . \quad . \quad (10)$$

subject to the condition that $x = \frac{\wedge}{2}$ when $y = \frac{L}{2}$

Hence, substituting in (10), $P = \frac{\wedge^2}{2L}$ . . . (11)

Now, x in the rectangular co-ordinates of the cam chart equals a_1 in the polar co-ordinates of the actual cam pitch curve, and y is the increase or decrease in ρ_1 above or below the radius R. Hence the function f in equation (8) for a uniform acceleration pitch curve may be written as

$$\rho_1 = R \pm \frac{a_1}{P} \quad . \quad . \quad . \quad . \quad (12)$$

where the upper and lower signs apply to the parabolas which extend from their vertices away from and toward the cam, respectively.

Supposing the following values known; the roller radius, r; the pitch inner concentric circle or outer concentric circle either represented by R; the lift, L; and the arc, $\wedge$, over which the complete uniform acceleration curve extends; then equations (7), (9), (11) and (12) give the solution of the complete uniform acceleration curve.

For the profile given at B of Figure 18, L and $\wedge$ are not known. The valve closure occurs at the point designated by A in Figure 20. This point corresponds to a large clearance, b, above the inner concentric circle of the complete uniform acceleration curve, and to an angle, a, ahead of the vertex of the inner concentric circle of the complete uniform acceleration curve. The values a and b are computed from the values of the allowable rate of closure of the valve at the point of closure necessary for taking up clearance variation errors, the given lift to the point of valve closure, and the given arc over which the closure of the valve extends.

Let m = slope of parabola at point of valve closure
= allowable rate of closure of valve at point of closure necessary to take up clearance variation errors.

l = lift to point of valve closure

λ = arc over which the closure of valve extends.

The values of a and b are desired in terms of l, λ, m.

Now, $\wedge = \lambda + a$, $L = l + b$ and substituting in (11).

$$P = \frac{(\lambda + a)^2}{2(l + b)}$$

From (10) $$\frac{dy}{dx} = m = \frac{2a}{P}$$

Hence $$m = \frac{4a\,(l + b)}{(\lambda + a)^2} \quad . \quad . \quad . \quad (13)$$

Substituting a, b and P in (10) we get

$$a^2 = \frac{(\lambda + a)^2 b}{2\,(l + b)} \quad . \quad . \quad . \quad . \quad (14)$$

Multiplying the two equations (13) and (14) together we obtain,

$$a = \frac{2b}{m} \quad . \quad . \quad . \quad . \quad (15)$$

Substituting a as given by (15) in equation (13) and solving the resulting expression for b, we obtain

$$b = \frac{m\lambda}{2} - l \pm \sqrt{\left(\frac{m\lambda}{2} - l\right)^2 + \frac{m^2\lambda^2}{4}} \quad . \quad . \quad (16)$$

We may also write from the above equations that

$$P = \frac{(\lambda + a)^2}{2(l + b)} = \frac{a^2}{b} = \frac{2a}{m} = \frac{4b}{m^2} \quad . \quad (17)$$

The analytic solution of the desired uniform acceleration cam profile consists therefore in solving the equations.

$$\rho^2 = \rho_1^2 + r^2 - \sqrt{\frac{4\rho_1^4 r^2}{\left(\frac{d\rho_1}{da_1}\right)^2 + \rho_1^2}} \quad . \quad . \quad (7)$$

where

$$\rho_1 = R \pm \frac{a_1^2}{P} \quad . \quad . \quad . \quad (12)$$

and

$$\frac{d\rho_1}{da_1} = \pm \frac{2a_1}{P}$$

and

$$\cos(a - a_1) = \frac{\rho^2 + \rho_1^2 - r^2}{2\rho\rho_1} \quad . \quad . \quad (9)$$

the condition for desired accuracy of delivery for small differences in clearances at the tappets being given by

$$P = \frac{(\lambda + a)^2}{2(l + b)} = \frac{a^2}{b} = \frac{2a}{m} = \frac{4b}{m^2} \quad . \quad (17)$$

where

$$b = \frac{m\lambda}{2} - l \pm \sqrt{\left(\frac{m\lambda}{2} - l\right)^2 + \left(\frac{m\lambda}{2}\right)^2} \quad . \quad (16)$$

and

$$a = \frac{2b}{m} \quad . \quad . \quad . \quad . \quad (15)$$

In the following pages is given an example of the application of these equations to the solution of the closing arc of a cam profile for which the following data is given.

l = lift = $\frac{1}{4}$ inches (as previously pointed out, this is the average lift of the Bureau Type suction valve).

λ = arc over which closure of valve extends = 85 degrees

r = roller radius = 1 inch

R = Radius on pitch curve corresponding to the vertex of the parabola whose curve extends from the vertex toward the cam

= outer concentric circle = 1.875 inches

m = slope of parabola at point of valve closure

= allowable rate of closure for taking up clearance variation errors of valve closure = .003 inch per degree.

On page 60 of the February, 1925, JOURNAL the valve travel to plunger travel ratio is given as 1:3.1875 and the stroke as 1.03125. Therefore, the rate of closure of valve with respect to crank travel (θ) in degrees is

$$\frac{\frac{1}{2}\text{ stroke}}{3.1875}\ \frac{d}{d\theta}\ \text{vers}\ \theta$$

$$= \frac{1.03125 \times \pi}{2 \times 3.1875 \times 180} \sin\theta$$

and for θ = 90 degrees or at full power, this equals .00282 inch per degree of pump crank travel.

Substituting in (16) we obtain.

$$b = \frac{.003 \times 85}{2} - \frac{1}{4} + \sqrt{\left(\frac{.003 \times 85}{2} - \frac{1}{4}\right)^2 + \left(\frac{.003 \times 85}{2}\right)^2}$$

$$= .0543120$$

Hence from (15) a = 36.2080

and from (17) P = 24,138.7

R^1 = radius on pitch curve corresponding to the vertex of the parabola which extends from its vertex away from the cam

$= R - (l + b) = 1.570688$

Therefore the equations of the pitch curve (12) become

$$\rho_1 = 1.570688 + .0000414273\ a_1^2 \quad . \quad (18)$$

and

$$\rho_1 = 1.875 - .0000414273\ a_1^2 \quad . \quad (19)$$

Equations (18) and (19) refer to the portion of the pitch curve profile obtained from the parabolas which extend from their vertices away from the cam and toward the cam, respectively. Angle a_1 in both curves is measured from the point corresponding to the vertex of the parabola considered. Equation (18) gives the inner half of the total uniform acceleration curve and (19) the outer half. Their point of tangency is at $\rho_1 = R - \frac{1}{2}(l + b)$, $a_1 = \frac{1}{2}(\lambda + a)$ which by substituting gives

$$\rho_1 = 1.722844,\ a_1 = 60.6040$$

The radius of the base circle of the pitch curve $= R - l$. The point on (18) corresponding to this is the point of valve closure or where $\rho_1 = R - l = R^1 + b = 1.625$ and

$$a_1 = a = 36.2080.$$

Any angle measured between the radius to this point and the radius to any point of the cam profile will be spoken of as the "cam angle". The cam profile to one side of this radius, OA, Figure 19 is the cam base circle whose radius is $\rho = R - l - r$ and also equal to $R^1 + b - r$ which by substituting the numerical values gives the cam base circle radius OB $= .625$. To the other side of this radius the cam profile to the point K is a circle whose radius is r and whose center is A. The remaining cam profile to the outer concentric circle is that corresponding to equations (18) and (19) for the inner and outer portions of the uniform acceleration pitch profile; equation (18) being used for all values of ρ_1 less than 1.722844 and (19) for all values of ρ_1 greater than that value.

Calculation for the coordinates ρ and a of the point K, Figure 19 is given in Figure 21, to illustrate the method used to obtain ρ and a from any value of a_1.

Accuracy to four decimals of an inch is desired as an ideal to be attained in construction. To obtain this, five placed tables were used and the calculation was carried out to six significant figures.

Step number 1 solves for ρ_1 from the given value of a_1, using equation (18). This corresponds to solving (12) of the set of general equations. The given value of a_1 (36.2080) is given within the parenthesis, its square (1,311.019) is given above it, the product (.054312) of the square times its coefficient is written below the constant (1.570688) in the equation for ρ_1 and the sum (1.625) of the latter two figures which is equal to ρ_1, is written after the equal sign. The value of $2\rho_1$ (3.25) which is used below is written above ρ_1. Next ρ_1^2 (2.640625) is written to the right of squared bracket and equal sign. And $2\rho_1^2$ (5.281250) is written above ρ_1^2.

Steps numbers 2, 3, 4, 5, 6 and 7 solve equation (7) of the set of general equations for ρ.

Steps numbers 8, 9 and 10 solve equation (9) for a.

Step number 11 gives the cam angle which in general for equation 18 is equal to $a - a$ and for equation (19) is equal to $\lambda - a$ or $\wedge - a - a$.

The value of ρ_1 for this particular value of a_1 was already known. Therefore, extra work was performed in step No. 1 in deriving it. For purposes of illustration, however, the general method was used to obtain ρ_1 from the given value of a_1 as this is the method to be used to obtain ρ_1 when this value corresponding to some value of a_1 is not known.

The point of tangency of the two parabolas and two intermediate points were then calculated. The results are given in table A of Figure 22. A curve of $a - a_1$ plotted on the corresponding cam angles is given in Figure 22. From this curve the approximate values given in table B are obtained

CAM PROFILE CALCULATION

$\alpha = 36{\cdot}208{,}0°$

STEP 1 $\rho_1 = 1{\cdot}570{,}688 + 10^{-6} \times 41{\cdot}427{,}3 \left(\frac{1311{\cdot}019}{(36{\cdot}208{,}0)^2}\right)^2 = \sqrt{1{\cdot}625}$ $2\rho = \frac{3{\cdot}25}{\alpha^2} = 2\rho = \frac{5{\cdot}281{,}250}{\alpha^2} = 2{\cdot}640{,}625$

STEP 2 $\frac{d\rho}{d\alpha_1} = 10^{-6} \times 82{\cdot}854{,}6 \times \frac{36{\cdot}208{,}0}{\alpha_1} = {\cdot}171{,}887 \Big]^2 = {\cdot}029{,}545$

STEP 3 $\rho_1^2 + \left(\frac{d\rho}{d\alpha_1}\right)^2$ SUM $= 2{\cdot}670{,}170$

STEP 4 $\rho_1^2 + r^2 = 3{\cdot}640{,}625$

STEP 5 $\sqrt{\frac{(2\rho_1^2)^2}{\left(\frac{d\rho_1}{d\alpha}\right)^2 + \rho_1^2}} = \sqrt{\frac{27{\cdot}891{,}60}{2{\cdot}670{,}170}} = \sqrt{10{\cdot}445{,}628} = 3{\cdot}231{,}970$

STEP 6 $\rho^2 = \text{DIFF.} = {\cdot}408{,}655$

STEP 7 $\rho = {\cdot}639{,}261$

STEP 8 $\cos(\alpha - \alpha_1) = \frac{\rho^2 + R^2 - r^2}{2R\rho} = \frac{2{\cdot}049{,}280}{2{\cdot}077{,}598} = {\cdot}986{,}369$

STEP 9 $\alpha - \alpha_1 = 9° 28' 22'' = 9{\cdot}470{,}3°$

STEP 10 $\alpha = 45{\cdot}678{,}3°$

STEP 11 CAM ANGLE $= 45{\cdot}678{,}3 - 36{\cdot}208{,}0 = 9{\cdot}470{,}3°$

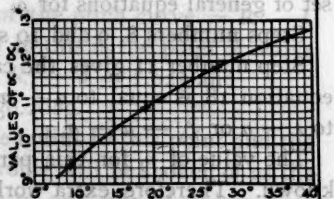
FIGURE 21

TABLE A

α_1	CAM ANGLE	$\alpha - \alpha_1$
36·2080	9·4703	9·4703
44·3400	18·9703	10·8383
52·4721	28·1405	11·8764
60·6041	37·0022	12·6061

TABLE B

CAM ANGLE	$\alpha - \alpha_1$ FROM CURVE	α = 36·2080 + CAM ANGLE	α_1
10	9·56	46·2080	36·648
11	9·72	47·2080	37·488
12	9·88	48·2080	38·328
13	10·02	49·2080	39·188
14	10·175	50·2080	40·033
15	10·31	51·2080	40·898
20	10·97	56·2080	45·238
25	11·55	61·2080	49·658
30	12·04	66·2080	54·168
35	12·47	71·2080	58·738



CURVES FOR APPROXIMATE DETERMINATION OF α, FOR VARIOUS CAM ANGLES IN SOLVING FOR THE CAM PROFILE CORRESPONDING TO THE PITCH CURVE AS GIVEN BY $\rho_1 = 1{\cdot}570{,}688 + 10^{-6} \times 41{\cdot}427{,}3\alpha^2$

FIGURE 22

for a_1 for equal angular values of the cam angles, of which every degree from 10 degrees to 15 degrees and every 5 degrees over the remaining arc is to be computed. A calculation with these approximate values similar to that given in Figure 20 is then made, and a correction applied to obtain a more correct value of the cam angle. This is illustrated in Figure 22 where ρ for the cam angle equal to 30 degrees is worked out. The approximate value of a_1 from Figure 21 is 54.168, which by calculation up to step 11 similar to that in Figure 20, gives the corresponding cam angle to be 30.014 degrees and $\rho = .719453$. The cam angle is .014 degree too large so that ρ should be decreased by a slight amount. Steps No. 12 to No. 15 give this correction, using the following reasoning:

Referring to Figure 19, draw DG tangent to the cam profile at D, and DH perpendicular to the radius OD. Then tan $\angle$ GDH is the rate of change of ρ with respect to $\rho\, a$ or lineal displacement on a concentric circle at this point, a being measured in radians, and $\frac{\pi}{180}$ tan $\angle$ GDH will be this rate of change when a is given in degrees. Now, since DH $\perp$ OD and DG $\perp$ CD, $\angle$ GDH is equal to 180 degrees $-$ $\angle$ CDO which is the external angle at D of the triangle CDO.

Hence $\angle$ GDH $=$ $\angle$ DCO $+$ $\angle$ COD.

$$= \tan^{-1}\left(\frac{d\,\rho_1}{d\,a_1}\cdot\frac{1}{\rho_1}\right) + a - a_1$$

Therefore, the ρ correction equals

$$\frac{\pi}{180}\tan\left[\tan^{-1}\left(\frac{d\,\rho_1}{d\,a_1}\cdot\frac{1}{\rho_1}\right) + a - a_1\right] \times \text{deviation of cam angle} \times \rho$$

The solution of ρ and the cam angle using equation (19) instead of (18), which was used in the above illustration, is similar to the calculations given above. A curve of $a - a_1$,

is plotted on cam angles, approximate values of a_1 are determined from this curve for equal spaced cam angles, ρ and the cam angle are then calculated from these approximate values, and a correction is applied at the end of the calculation to correct ρ for the deviation of the cam angle from the value desired.

As explained previously the cam profile over the arc BK of Figure 19 is a circle whose radius is r and whose center is A. The cam angle and ρ for point K as given in Figure 20, are 9.4703 degrees and .639261 seconds respectively. The values for ρ corresponding to cam angles for every degree from 0 degrees to 9 degrees inclusive, may be solved for as follows:

The equation of this circle in polar co-ordinates is

$$\rho^2 + \rho_1^2 - 2\,\rho\rho_1 \cos(\varphi - \varphi_1) - r^2 = 0$$

where (ρ, φ) is any point on the circle and (ρ_1, φ_1) and r are its center and radius respectively.

Assume that the radius vector through the center of the circle is the initial line and that φ is the cam angle. Then putting $r = 1$ and $\rho_1 = 1.625$, we get

$$\left(\frac{\rho}{1.625}\right)^2 - 2\left(\frac{\rho}{1.625}\right)\cos\varphi + 1 - \frac{1}{(1.625)^2} = 0$$

so that

$$= 1.625\,(\cos\varphi \pm \sqrt{\cos^2\varphi - .621302})$$

It is therefore a simple matter to solve for ρ for any value of φ

Thus, let $\varphi = 6$ degrees; $\cos\varphi = .9945218$

$$\cos^2\varphi = .989074$$
$$\cos^2\varphi - .621302 = .367772$$
$$\sqrt{.367772} = .606442$$
$$\cos\varphi - .606442 = .388080$$
$$\therefore \rho = 1.625 \times .388080 = .630630$$

The cam profile whose radial distances are given in tabular form in Figure 24 and in graphical form in Figure 25, is the result of continuing the calculations by the methods given.

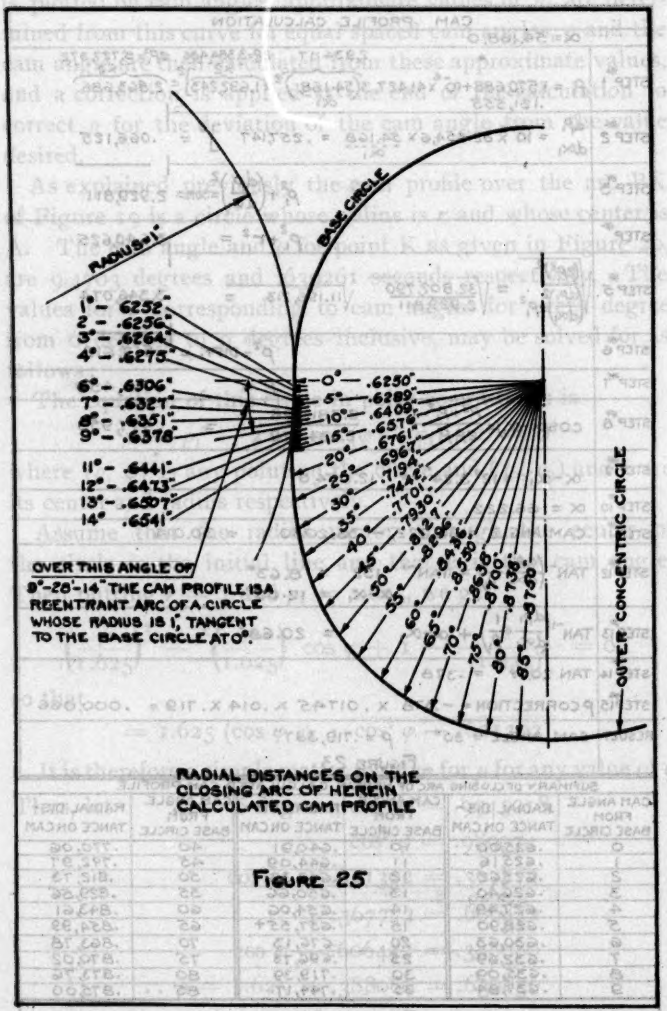
CAM PROFILE CALCULATION		
	α = 54.168,0	
STEP 1	$\rho_1 = 1.570,688 + 10^{-6} \times 41.427.3 \left(\frac{54.168}{\alpha_1}\right)^2 = 1.692,243$ (2,934.17 / x2; 2P = 3,384,486; 2P² = 5,727,372 / x2)	$= 2.863,686$
STEP 2	$\frac{d\rho_1}{d\alpha_1} = 10^{-6} \times 82.854,6 \times \frac{54.168}{\alpha_1} = .257,147$	$\Big]^2 = .066,125$
STEP 3	$\rho_1^2 + \left(\frac{d\rho_1}{d\alpha_1}\right)^2$ = SUM = 2.929,811	
STEP	$\rho_1^2 + r^2 =$	3.640,625
STEP 5	$\sqrt{\left(\frac{d\rho_1}{d\alpha_1}\right)^2 + \rho_1^2} = \sqrt{\frac{32,802,790}{2.929,811}}$	$\sqrt{11.196,213} = 3.346,074$
STEP 6	ρ^2 - DIFF. =	.517,612
STEP 7	ρ =	.719,453
STEP 8	$\cos(\chi - \alpha_1) = \frac{\rho^2 + \rho_1^2 - r^2}{2\rho_1\rho} = \frac{2.381,298}{2.434,979}$	$= .977,953$
STEP 9	$\alpha - \alpha_1$ = 12° 3.24′ =	12.054,0°
STEP 10	α = 66.222	
STEP 11	CAM ANGLE = 66.222, − 36.2080	= 30.014
STEP 12	$\text{TAN}^{-1}\left(\frac{d\rho_1}{d\alpha_1} \cdot \frac{1}{\rho_1}\right) = \text{TAN}^{-1}$ .152 = 8.63° $\alpha - \alpha_1$ = 12.05°	
STEP 13	$\text{TAN}^{-1}\left(\frac{d\rho_1}{d\alpha_1} \cdot \frac{1}{\rho_1}\right) + \alpha - \alpha_1$	= 20.68°
STEP 14	TAN 20.68° = .378	
STEP 15	ρ CORRECTION = − .378 X .01745 X .014 X .719 =	.000,066
RESULT	CAM ANGLE = 30°	ρ = .719,387"

FIGURE 23

SUMMARY OF CLOSING ARC OF HEREIN CALCULATED CAM PROFILE

CAM ANGLE FROM BASE CIRCLE	RADIAL DISTANCE ON CAM	CAM ANGLE FROM BASE CIRCLE	RADIAL DISTANCE ON CAM	CAM ANGLE FROM BASE CIRCLE	RADIAL DISTANCE ON CAM
0	.625,00	10	.640,91	40	.770,06
1	.625,16	11	.644,09	45	.792,97
2	.625,62	12	.647,33	50	.812,73
3	.626,40	13	.650,66	55	.829,56
4	.627,49	14	.654,06	60	.843,61
5	.628,90	15	.657,55+	65	.854,99
6	.630,63	20	.676,13	70	.863,78
7	.632,69	25	.696,73	75	.870,02
8	.635,09	30	.719,39	80	.873,76
9	.637,84	35	.744,17	85	.875,00

FIGURE 24



RADIAL DISTANCES ON THE CLOSING ARC OF HEREIN CALCULATED CAM PROFILE

FIGURE 25

Theoretically, a slight difference from actual uniform acceleration will result from the angularity of the rocket arm which actuates the valve tappet. Assume the length of the rocker is $3\frac{1}{4}$ inches. This will give an angularity from the center of the roller's travel to the end equal to

$$\sin^{-1}\left(\frac{\frac{1}{2}\text{ Lift}}{\text{Rocker Arm Length}}\right) = \sin^{-1}\frac{.125}{3.25} =$$

$$\sin^{-1} .03846 = 2 \text{ degrees } 12.25 \text{ minutes}$$

If the chord of the roller's arc of motion be considered the desired line of motion, the maximum divergence from this straight line motion will occur at one half the lift and will be equal to $3.25 \times$ vers 2 degrees 12.25 minutes $= 3.25 \times .000742 = .00241$ seconds. Now, by a calculation similar to that given in Figure 23, we obtain at the cam angle 45 degrees, $\frac{d\,\rho_1}{d\,a_1} \cdot \frac{1}{\rho_1} = .1325$ which is the rate of change of the pitch curve radius at approximately the center of the lift per lineal displacement on a concentric circle at this point. Therefore, the divergence of .00241 seconds from straight line motion corresponds to a divergence from the curve for uniform acceleration for straight line motion of approximately $.00241 \times .1325$ or .00032 seconds. The error accumulates from the position of zero cam angle to the position at one-half the lift. This is approximately 45 degrees of cam angle or over 9 radial distances 5 degrees apart, and therefore corresponds to an error of $\frac{.00032}{9}$ or .0000356 seconds between successive radial distances. Hence, the error due to the angularity of the rocker arm is negligible, compared to the desired accuracy.

Calculations for the per cent of delivery unbalance which would obtain from the use of the above calculated cam profile are given in Figure 26. Clearance errors of .001, .002, .003 and .004 of an inch are considered, and the resulting per cent

LET ϕ = THE ANGLE IN DEGREES BETWEEN THE RADII TO THE POINT CORRESPONDING TO THEORITICAL VALVE CLOSURE ON THE PITCH CURVE AND ANY POINT ON THAT CURVE. THIS ANGLE IS THE ANGLE AOC IN FIGURE 19.

THEN $\alpha_1 = \phi + \alpha$
$= \phi + 36.208,0$

SUBSTITUTING IN EQUATION 18, THIS GIVES

$$\rho_1 = 1.570,688 + 10^{-6} \times 41.427,3\,(\phi + 36.208,0)^2$$

OR,

$$\phi = -36.208,0 \pm \sqrt{\frac{\rho_1 - 1.570,688}{10^{-6} \times 41.427,3}}$$

LET $\rho_1 = \rho_1' + \delta\rho_1$ WHERE ρ_1' IS THE RADIUS CORRESPONDING TO $\alpha_1 = \alpha$.

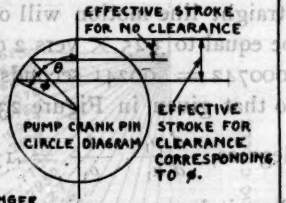
THEN $\rho_1 = 1.625 + \delta\rho_1$

AND $$\phi = -36.208,0 + \sqrt{\frac{\delta\rho_1 + .054,312}{10^{-6} \times 41.427,3}}$$

IF $\delta\rho_1$ IS THE CLEARANCE ERROR, ϕ FROM THE ABOVE FORMULA GIVES THE CAM ROTATION TO TAKE UP THIS CLEARANCE.

VALUES OF ϕ FOR CLEARANCE ERRORS

① $\delta\rho_1$	② $\delta\rho$ +36.2080	③ ②$\times 10^4$ +41.4273	④ $\sqrt{③}$	ϕ = ④ − 36.208,0
0	.054,312	1311.02	36.208,0	0
.001"	.055,312	1335.15	36.539,7	.3317° OR 19'.91
.002"	.056,312	1359.29	36.8686	.6606° OR 39'.64
.003"	.057,312	1383.43	37.1945	.9865° OR 59'.19
.004"	.058,312	1407.57	37.5176	1.3096° OR 1°18'.58



FULL POWER = $\frac{1}{2}$ STROKE EFFECTIVE PLUNGER TRAVEL $\theta = 90°$

$\frac{2}{3}$ POWER = $\frac{2}{3}$ FULL POWER EFFECTIVE PLUNGER TRAVEL $\theta = 70°\text{-}31'.74$

$\frac{1}{3}$ POWER = $\frac{1}{3}$ FULL POWER EFFECTIVE PLUNGER TRAVEL $\theta = 48\text{-}11'.36$

IDLING = 15% FULL POWER EFFECTIVE PLUNGER TRAVEL $\theta = 31°\text{-}47'.31$

$$\text{PERCENT ERROR} = 100\left(\frac{\text{VERS}(\theta+\phi)}{\text{VERS}\,\theta} - 1\right)$$

SOLUTION, FOR ERROR DUE TO CLEARANCE ERRORS

	θ	VERSθ	$\delta\rho_1$ = .001"		$\delta\rho_1$ = .002"		$\delta\rho_1$ = .003"		$\delta\rho_1$ = .004"	
			θ + φ	VERS(θ+φ)	θ + φ	VERS(θ+φ)	θ + φ	VERS(θ+φ)	θ + φ	VERS(θ+φ)
FULL POWER	90°	1.	90° 19'.91	1.00579	90° 39'.64	1.01153	90° 59'.19	1.01722	91° 18'.58	1.02286
⅔ I.H.P.	70° 31'.74	.6667	70° 51'.65	.67214	71° 11'.38	.67756	71° 30'.93	.68295	71° 51'.32	.68830
⅓ I.H.P.	48° 11'.36	.33333	48° 31'.27	.33766	48° 51'.00	.34197	49° 10'.55	.34626	49° 29'.99	.35054
IDLING	31° 47'.31	.15000	32° 7'.22	.15309	32° 26'.95	.15613	32° 46'.54	.15920	33° 5'.89	.16226

	VERS(θ+φ) ÷ VERSθ				PERCENT ERROR			
CLEARANCE	.001	.002	.003	.004	.001	.002	.003	004
FULL POWER	1.0058	1.0116	1.0172	1.0230	.58	1.16	1.72	2.30
⅔ I.H.P.	1.0082	1.0163	1.0244	1.0324	.82	1.63	2 44	3.24
⅓ I.H.P.	1.0130	1.0259	1.0388	1.0510	1.30	2.59	3.88	5.16
IDLING	1.0206	1.0409	1.0613	1.0817	2.06	4.09	6.13	8.17

Figure 26

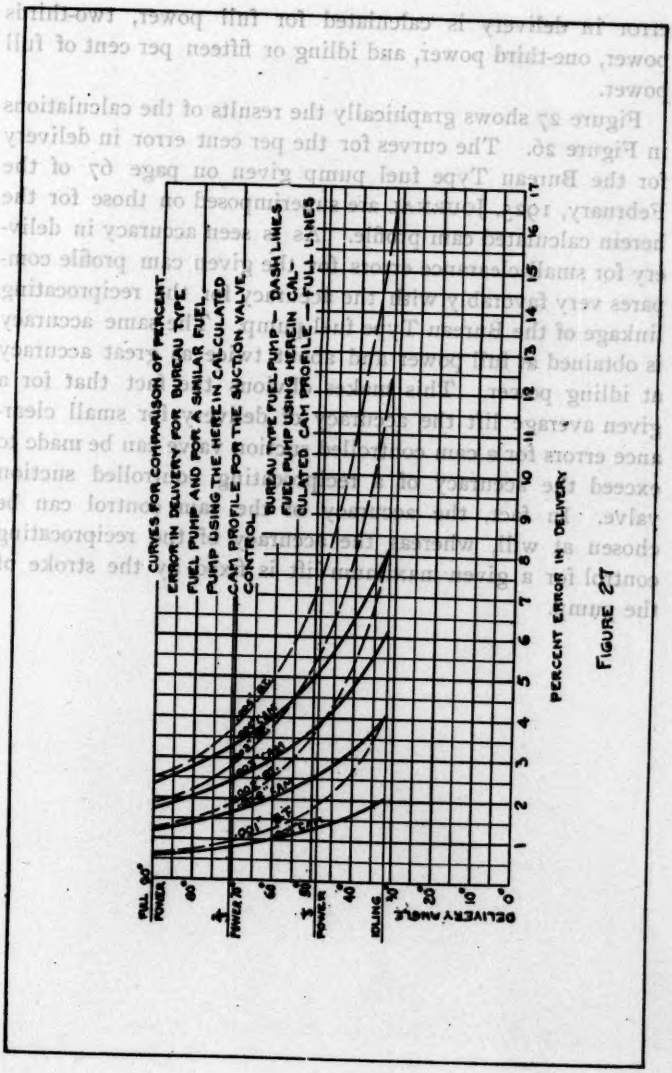
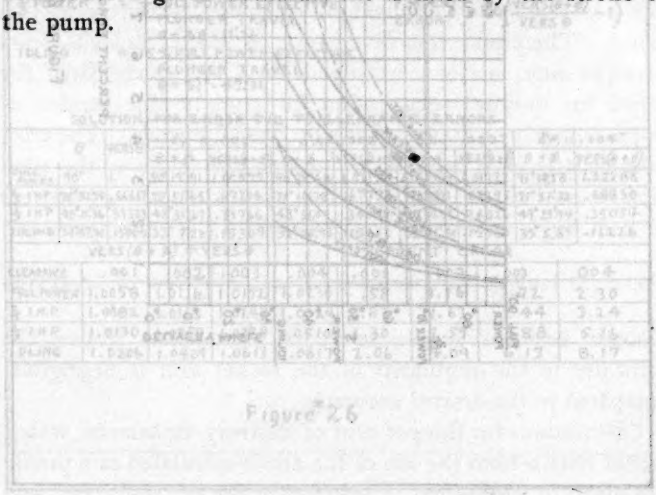
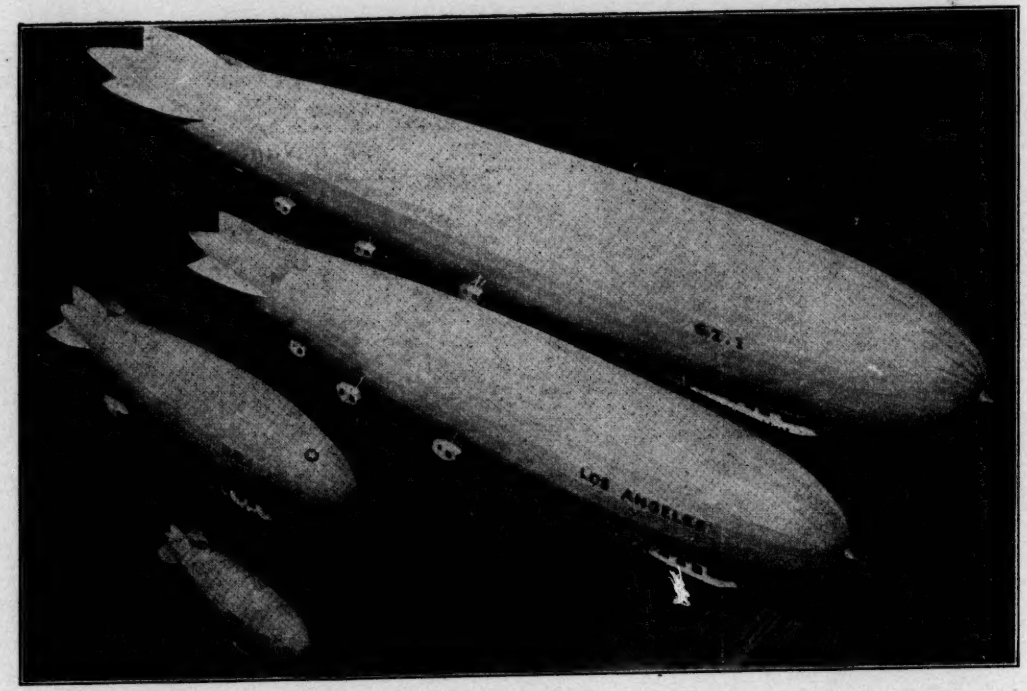

CURVES FOR COMPARISON OF PERCENT ERROR IN DELIVERY FOR BUREAU TYPE FUEL PUMP AND FOR A SIMILAR FUEL PUMP USING THE HEREIN CALCULATED CAM PROFILE FOR THE SUCTION VALVE CONTROL
BUREAU TYPE FUEL PUMP — DASH LINES
FUEL PUMP USING HEREIN CALCULATED CAM PROFILE — FULL LINES
FULL POWER
¾ POWER
½ POWER
IDLING
DELIVERY ANGLE
PERCENT ERROR IN DELIVERY

FIGURE 27

error in delivery is calculated for full power, two-thirds power, one-third power, and idling or fifteen per cent of full power.

Figure 27 shows graphically the results of the calculations in Figure 26. The curves for the per cent error in delivery for the Bureau Type fuel pump given on page 67 of the February, 1925, JOURNAL are superimposed on those for the herein calculated cam profile. As is seen accuracy in delivery for small clearance errors for the given cam profile compares very favorably with the accuracy for the reciprocating linkage of the Bureau Type fuel pump. The same accuracy is obtained at full power and about twice as great accuracy at idling power. This makes obvious the fact that for a given average lift the accuracy of delivery for small clearance errors for a cam controlled suction valve can be made to exceed the accuracy of a reciprocating controlled suction valve. In fact, the accuracy of the cam control can be chosen at will, whereas the accuracy of the reciprocating control for a given maximum lift is fixed by the stroke of the pump.





THE NEXT STEP.

The illustration shows the smallest and the largest airship, as they will appear if and when completed, in comparison with the United States naval airship *Los Angeles* (*Z.R.-3*) and the U. S. Army's semi-rigid air cruiser *R.S.-1*. The *Pony Blimp*, in the immediate foreground, is now under way in the aeronautical workshops of the Goodyear Co. in Akron. This non-rigid dirigible will be 100 feet long and will hold 47,000 cubic feet of gas. The enclosed cabin will have accommodations for three passengers and a pilot. The *G.Z.-1* is now being designed in Akron, Ohio, by the Goodyear-Zeppelin Corp. This mammoth air liner, which is still only a project, is to be more than 900 feet in length and will hold over 5,000,000 cubic feet of gas.

NOTES.

THE PROGRESS OF AERONAUTICS.

Fifth Annual Report of Aeronautical Research Committee.

The report of the Aeronautical Research Committee for the year 1924-1925 has just been published and contains a great deal of interesting matter. It is pointed out that at the end of the financial year a reconstitution of the Committee took place, the new terms of reference being as follows:—

"(1) To advise the Secretary of State on scientific problems relating to aeronautics. (2) To make from time to time recommendations to the Air Council as to any researches which the Committee consider it desirable to initiate, and as to any matters referred to them by the Council. (3) To supervise the aeronautical researches at the National Physical Laboratory initiated by them, and, if requested to do so by the Air Council, any other researches connected with Aeronautics. (4) To make an annual report to the Air Council of the research work which the Committee consider should be undertaken at the National Physical Laboratory, or elsewhere, together with an estimate of expenditure at the National Physical Laboratory. (5) To investigate the causes of such accidents as may be referred to them by the Air Council, and to make recommendations as to the prevention of accidents in the future. (6) To promote education in aeronautics by cooperating with the Governors of the Imperial College and in any other way within their power. (7) To assist with advice any research carried out by or on behalf of the Aeronautical Industry, and to make available any information of value to the Industry so far as is compatible with public interests. (8) To make an annual report to the Secretary of State for Air."

In the previous terms of reference the Committee were instructed to advise on scientific and technical problems relating to the construction and navigation of aircraft. This change, it is pointed out, has been made possible by the reorganization at the Air Ministry at that establishment, which formerly dealt with technical development and research and as a further consequence the Air Council has decided that future membership shall be confined solely to members appointed in virtue of their scientific standing, rather than as representatives of definite interests.

A. R. C. AND AIRCRAFT INDUSTRY.

The report points out that contact between the Committee and the Aircraft Industry will still be maintained by the following arrangements:—"(a) Annually, before the program of research for the ensuing year is decided upon, a joint meeting of the Aeronautical Research Committee and the Industry will take place, at which the carrying out of particular researches during the ensuing year will be freely discussed. After this joint meeting has been held, the Aeronautical Research Committee and the Society of British Aircraft Constructors will forward their respective proposals for research to the Air Council, who will decide upon the program for the ensuing year. A copy of the decisions will then be forwarded to the Aeronautical Research Committee and the Society of British Aircraft Constructors. (b) Six months after the program has been decided upon a second joint meeting of the Aeronautical Research Committee and the Society of British Aircraft Constructors will take place, at which the progress of the last six months

will be reported and a discussion will ensue. As a result of this second meeting, the Air Council will decide if any modifications of the annual program are necessary.

The Committee welcomes the grant of £1500 made available by arrangement with the Air Ministry for the purpose of grants to individual research workers. The amount of this grant exceeded by £500 that allowed in the previous year, in order that funds might be made available for special experiments on light aeroplanes. Grants have also been made for work on experimental research on redundant structures, on wind tunnel and theoretical research on skin friction, and on theoretical research on the problem of control at low speeds.

The report states that in addition to the provision of the above grants, some work financed by the Department of Scientific and Industrial Research "has been carried out on elasticity and fatigue at Oxford University" (this is the official wording!), and a program of investigation on single cylinder internal-combustion units is in hand at the Universities of Cambridge, Durham and Manchester.

The scheme of the Sub-Committees and Panels directing the work in detail is under consideration by the reconstituted Committee. It is stated that during the past year no new problems have been referred to the Meteorology Sub-Committee or to the Fire Prevention Committee, and these Sub-Committees have consequently not met. Under the new terms of reference the Air Inventions Sub-Committee will no longer exist, the work being carried out in its entirety by the Air Inventions Department of the Air Ministry.

Reference is also made in the report to the work of the Airship Co-ordinating and the Structures Sub-Committees.

The Committee point out that the new arrangements to which reference was made last year have resulted in the publications of the Committee being issued more rapidly and brought up to date. Not only are the individual Reports and Memoranda issued within a few weeks of their approval by the Aeronautical Research Committee, but the annual volumes of collected technical papers for past years are now on sale.

Equipment for Experimental Work—The Committee did not make any new proposals this year for increasing the equipment at the National Physical Laboratory, except in connection with certain proposals for a new high-speed tank. It is considered important that approval for the construction of this tank should be given, as with the progress in seaplane development it is imperative that equipment should exist by means of which comparative tests may be carried out at speeds considerably greater than those now possible with the existing equipment at the William Froude Tank, and it is thought that better progress in seaplane development will result therefrom as no means exist at present for making model experiments on a high-speed seaplane.

Progress of Research—The report points out that in two directions a substantial stage in progress has been reached. One of these is the problem of stalled flight, which with the completion of the first flight experiments on the slot-and-aileron control, is now emerging from the pioneer stage. The other promising advance has been made in the course of investigation on elasticity and fatigue.

AERODYNAMICS.

Research upon the control of stalled aeroplanes has gone steadily forward, and the conclusion is formed that conventional aeroplanes when stalled are defective in two respects—they have insufficient rudder power, and the ailerons when applied cause the aeroplane to turn and, by doing so, neutralize their direct effect on roll. It has been found that either an increase of

rudder power or the use of an aileron which does not turn the aeroplane would give the pilot power to regain an even keel, and so prevent the fatal spinning dive. Several devices have been tried, the most successful to date being a combination of a Handley Page slot on the leading edge, operated by a simple cam mechanism, in conjunction with a balanced aileron of the Bristol-Frisé type. This combination has been fitted to a standard Avro with the result that its control in the stalled state is greatly improved. In particular it is found that the spin, either in its initial stages or when fully developed, can be stopped by the ailerons without using the rudder.

For the purpose of further study of the behaviour of stalled aeroplanes, three special research aeroplanes are being built, and for use with these special aeroplanes, and for other researches on aeroplanes in flight, it is considered essential that an adequate supply of instruments should be forthcoming. Many satisfactory instruments have, it is stated, been developed by the R.A.E., but the report points out that hardly more than one complete set to include every instrument that might be needed has yet been constructed, and the Committee recommend "that the Air Ministry should take steps at once to increase the number of such instruments available, for example, by introducing a procedure whereby all instruments are placed in an instrument maker's hands for development on a production basis as soon as each has passed through the experimental stage."

Aerofoil Research—The Design Panel are continuing their investigation into scale-effect, and two new sections have been chosen, R.A.F. 26 and 31, for further comparative work between model and full scale. A theory of the design of aerofoils has been put forward by the R.A.E., and some six sections have been chosen on this basis with the object of obtaining in the first instance a good thick wing, and, secondly, a good racing wing. "The essential feature of the design," the report states, "was to curve the center line of a good symmetrical section into a circular arc of suitable camber, and in the case of high camber a cubic curve was also tried for the center line in order to reduce the movement of the center of pressure. The theoretical basis has been fully confirmed by the experimental results, and the most promising, R.A.F. 26, having a k_L max. of 0.460, k_D min. of 0.0047 and (k_L/k_D) max. of 22.9, has been chosen for further scale-effect research. Another section of the same series, R.A.F. 31, having a k_L max. of 0.537, k_D min. of 0.0060 and (k_L/k_D) max. of 20.8, will be used in the same connection. R.A.F. 26 is considered to be typical of racing sections and comparable with A.D. 1 and the Sloane sections, whereas R.A.F. 31 is considered to be typical of thick sections in use in present-day design, and has about the same lift and drag characteristics as R.A.F. 15. The aerofoil R.A.F. 25 was designed with a view to having a very low minimum drag, which at an l_v of 33 was found to be 0.0037; the maximum lift coefficient is 0.427, giving a ratio of minimum drag to maximum lift which shows a distinct improvement over R.A.F. 15. The last of the series, R.A.F. 33, has a reflexed trailing edge, and the results compare well with other reflexed wings.

"Attention has been drawn to the importance of knowing the characteristics of thick aerofoils, and the data available have been summarized and published in R. and M. 927. It is of interest to note that low minimum drags are obtainable with certain thick aerofoils, and of special interest in this connection is the German section 420 which has been tested at the N.P.L."

A new method for the measurement of aeroplane performance has been developed by an independent investigator, Capt. G. T. R. Hill, and it is hoped that an improvement in performance analysis will result, since by the use of this method no tests need be made without the air-screw running.

Of increasing importance is the problem of wing flutter, and the report states that this has been discussed with representatives of a number of firms

and that a preliminary theoretical attack has been made on the problem, which, it is pointed out, may need a large amount of experimental inquiry before a complete solution is obtained. Information regarding the rigidity of wings is being collected by the Airworthiness Department of the Air Ministry, and a series of accidents associated with flutter is being investigated by the Accident Sub-Committee.

A paragraph in the report states that "The Committee have viewed with interest a film of a flying appliance employing a rotating windmill, and the problems suggested by this new form of heavier-than-air craft are being attacked. Another new matter brought before the Committee deals with the aerodynamics of a rotating cylinder, and, in order to ascertain its possible usefulness on an aeroplane, experiments on a large scale and at a high speed are to be made in a wind tunnel."

SEAPLANES.

Important model experiments have been made at the National Physical Laboratory to ascertain the effect of moving the maximum beam of a seaplane to various fore and aft positions, and the results have shown that it should be possible to move the position of the maximum beam forward and shape the body aft so as to give a lighter tail and enable the amount of weight carried in the fore part of many seaplanes to be reduced.

The water absorption of seaplane hulls is receiving attention and a lengthy program of absorption experiments is in hand at the National Physical Laboratory. The program for next year will include experiments on cleanness of running, on the efficiency of the fin, on certain amphibian problems, on the scale effect of porpoising, and on the scale effect on small models with a view to ascertaining how small the model may be and yet give satisfactory results.

AIRSHIPS.

Wind tunnel work is stated to be in progress on airship models in connection with the design of new airships by the Cardington staff of the Air Ministry and by the Airship Guarantee Company. In view of the re-commencement of airship flying in this country a special Panel was appointed, which reported in 1924, and this report, R. & M. 970, has recently been published.

AERO ENGINES.

A large number of aero engine problems are under investigation, among which mention may be made of the problem of detonation, which is being steadily attacked, and owing to the urgency of the problem the work on anti-detonation substances is being given preference at the Air Ministry Laboratory. Certain experiments on the Ricardo variable-compression units, following approved programs of research, are in hand at Cambridge and Durham Universities.

The Engine Sub-Committee are giving considerable attention to problems in connection with increasing the power-weight ratio of engines, particularly at great heights. Pending the development of super-charged engines, the Committee have formed a high opinion of the bi-fuel system submitted by the Bristol Aeroplane Company in conjunction with a high-compression engine, and the report adds that additional efficiency at great heights can be obtained by the employment of a variable valve-timing gear.

On the question of technical officers the following interesting statement is found in the report: "The Committee call attention to the need of officers with more technical knowledge, and to the hope expressed in their last report that it will not be impossible to provide for highly skilled persons to make a direct technical diagnosis of each power-plant failure immediately it occurs, and to report it in a manner which will enable the Ministry to take

measures to prevent recurrence. They consider it desirable to utilize to the full the knowledge of such officers as possess the necessary technical qualifications."

AIR TRANSPORT.

In a supplement to the report it is stated that the Air Ministry Sub-Committee have met nine times during the past year, and have also visited Croydon in order to become acquainted with the steps taken to insure reliability of service. It is stated that the main measuring instrument used at present on the air transport service is the revolution recorder on the engine, which provides some information as to whether the engine has run satisfactorily. The Sub-Committee expresses the opinion that recording petrol flowmeters would not only amplify this information, but would indicate any serious waste of petrol.

On the question of engine reliability it has been ascertained that nothing is known of the correct "commercial rating" of an aeroplane engine, and the opinion is expressed that it is possible that the safe interval between overhauls could be varied if the percentage at which the engines are run were altered. For instance, it is considered certain that the life of an engine would be greatly increased if it were normally run at about 70 per cent of its power. The subject of three-engined aeroplanes has received attention and the report states that it is a matter of arithmetic to show that the reliability of such an aeroplane is greatly increased over the more usual type.

On the subject of landing in fogs it is interesting to find the following statement: "It would appear that the problem of alighting on an aerodrome under all conditions is near solution. By wireless direction, a pilot should find the vicinity of the aerodrome; the leader cable, which is developing satisfactorily, will enable him to determine the exact position of the aerodrome and his height above it, and the Neon lights should be adequate to help him in making the actual landing."

Economic Flight.—The last paragraph of the supplement to the report deals with the important question of economic flight, and states that an important investigation has been put in hand in order to ascertain the most economic distribution of structure for a three-engined aeroplane. Other subjects under investigation include the determination of the optimum flight speed for freight-carrying machines, the effect of head wind, and a consideration of the most economic length of stage of various sizes of aeroplanes.

Reference is made to the problem of refueling during flight so as to avoid the necessity for intermediate landings on long stages, and it is stated that experiments carried out at the R.A.E. have shown that the necessary maneuver can readily be carried out.

Finally, the question of the use of the sleeve-valve principle in its application to aero engines is briefly referred to.

Copies of the Report of the Aeronautical Research Committee for the year 1924-25 may be obtained from His Majesty's Stationery Office, the price being 1s. 6d. net.—"Flight," Aug. 27, 1925.

DUTCH AIRCRAFT CONSTRUCTION.

SOME IMPRESSIONS OF A VISIT TO THE FOKKER WORKS AT AMSTERDAM.

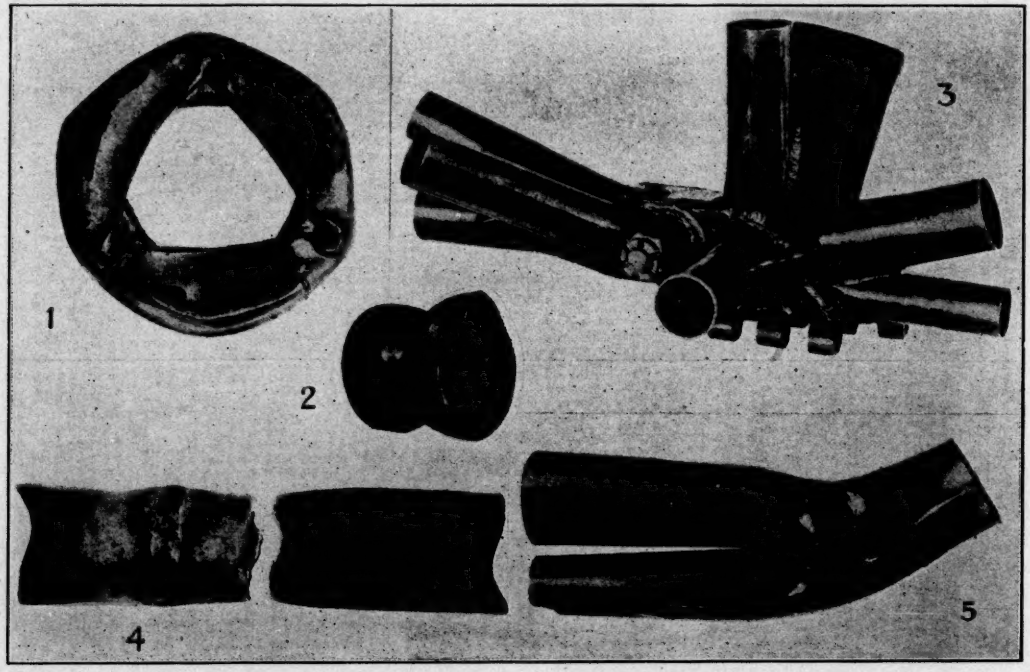
No matter how much one sees of machines at aerodromes and in the air, one never obtains a really intimate conception of their manufacture and construction until one has seen the same machines going through the works. This fact was strongly borne in upon us some time ago, when an opportunity offered of paying a visit to the famous Fokker works in Amsterdam. The great Dutch designer himself was not, unfortunately, at home, being at the

time absent on a visit to the United States. We had, however, a very excellent guide in Mr. B. Stephan, who is General Manager of the Fokker factories in Holland, and who speaks English so perfectly, that one is apt to forget, when going through the Fokker works in his company, that one is not in a British aircraft factory.

The present Fokker works, as some of our readers may be aware, are formed partly by two large buildings erected to house the machines exhibited at the Elta Show in 1919, and when these buildings were erected they were scarcely intended as permanent structures. For the time being, however, they have proved very convenient for the Fokker works, or to give them their proper title, the *N. V. Nederlandsche Vliegtuigenfabriek*, but it will probably not be long before the main works are transferred elsewhere. Already the firm has another factory at Veere, where most of the wing construction is carried out and the present Fokker factory at Paparverweg is mainly employed on experimental work. Add to this the fact that the Schiphol aerodrome is on the opposite side of Amsterdam from the present Fokker works, and it will be realized that this is scarcely an ideal arrangement. When Fokker took over the Elta buildings after the closing of the Show in 1919, there was considerable uncertainty as to whether any orders would be forthcoming, and it was not thought wise to launch out into heavy expenses for permanent works that might be without orders. Since that time, however, the factories have become very busy, supplying aircraft to all parts of the world. The present arrangement is now regarded more in the nature of a necessary evil, until better accommodation can be found. This we understand may be announced at any moment now, when it is hoped that a new Fokker factory will house all its departments together under one roof.

On the day of our visit to the Fokker works but a relatively small number of machines were to be seen, as it so happened that a batch had just been delivered, while most of the experimental machines completed were out on the Schiphol aerodrome. Sufficient could, however, be seen to indicate that, in spite of their somewhat temporary character, even the present works would be capable of quantity production on a not inconsiderable scale. On the normal work going on in the Fokker factory it is not proposed to dwell here, but rather will we turn to the experimental and test departments, in which was to be seen much that was of great interest.

Our visit to the Fokker works helped to solve a question which had often puzzled us—namely, how in constructing an experimental type of fuselage, the number of tubes running at various angles were cut to size and fitted. We had known, of course, that the Fokker method of welded-steel tube construction was claimed to be very simple and very adaptable, but we had been under the impression that, although this might apply to construction on a quantity-production basis, it would be a rather different story as regards the building of individual experimental machines. It was discovered, however, that even this kind of work is carried out with the greatest ease in the Fokker factory. It is, obviously, impossible for us to devote sufficient space to go into great detail, but, briefly, the system employed is as follows:—When a new fuselage is being produced, full-size drawings are got out, from which the size and length of tubes to be employed in any particular position are measured off direct. The workman places a length of tubing on his full-size drawing and marks off the required length. If the tube does not form a right angle with the other tube to which it is to be joined, the workman marks off the angle on the actual tube and then makes a small cut with a hacksaw. With a pair of ordinary tinsmith's shears he then cuts round the marking on the tube with the greatest of ease, and any slight further fitting which is required may be completed by the aid of a file. In erecting the fuselage the two tubes that are to be joined together are held in position, relative to one another, by a workman, while the welder puts on two or three



SOME EXAMPLES OF FOKKER WELDING: 1 AND 2 ILLUSTRATE WHAT WELDED TUBING LOOKS LIKE AFTER UNDERGOING COMPRESSION TESTS. 3. A TYPICAL FOKKER WELDED TUBULAR STEEL JOINT WHERE EIGHT MEMBERS MEET, NOT COUNTING THE UNDERCARRIAGE STRUTS, WHICH GO TO THE HINGED FITTING SHOWN UNDERNEATH. 4. A PIECE OF TUBING WITH WELDED BUT JOINT, AFTER HAVING UNDERGONE TENSION TESTS. 5. THIS WAS ORIGINALLY A WELDED "T" JOINT; THIS WELD HAS BEEN TESTED BY BENDING ONE TUBE THROUGH AN ANGLE OF APPROXIMATELY 90 DEGREES.

spot-welds, which are just sufficient to hold the tubes in the required position. Should the angle not be quite correct, or should one tube be slightly out of its proper plane, it is a very easy matter for the welder to change the angle, since there is sufficient elasticity in the steel wall of the tube and of the spot-welds to give to some considerable extent. When the tube has thus been lightly pulled into its proper position the weld is completed. The process sounds amazingly crude, but in actual fact it is extremely effective, and it would, indeed, be difficult to imagine anything more simple than the welding process as employed in the Fokker works. From the foregoing it will be realized that the task of completing a new fuselage, *i.e.*, one of different proportions, is an extremely simple one, and one begins to realize how it is that Fokker can turn out such a surprisingly large number of new types. The bugbear of premature standardization is entirely eliminated by his method, and a fuselage larger or smaller than any existing type can be produced in this manner in a very short time.

When a machine has been put into quantity production a somewhat different system is employed, and jigs are then constructed on which the fuselages are built, the jigs holding the four longerons in place while the struts are being welded to them. The work progresses with surprising speed, and Mr. Stephan informed us that it only took something like a fortnight to discover whether a man or woman had any aptitude for welding. At the end of a fortnight's welding it was quite obvious whether the person was ever likely to make a good welder or not, and for quantity-production, at any rate, the necessary skill, which is really very small, is readily acquired. Unless by the end of a fortnight the apprentice can make reasonably good welded joints, it is nearly always safe to assume that he never will make a good welder.

On the questions of repairs we were somewhat curious as to how these are effected, but even here the simplicity of the Fokker method was obvious. In this connection we cannot refrain from briefly referring to an experience of Mr. Stephan while he was in the Technical section of the Royal Dutch Air Service. Mr. Stephan frankly admits that he was somewhat doubtful of the merits of the Fokker welded-steel tube construction, but when some machines used by the Royal Dutch Air Service were damaged at the Air Service station, he was faced with the task of repairing them. At the time he knew nothing very much about how to tackle the job, and he put it off as long as he decently could, but at last the time came when something had to be done with the machines. The fabric was stripped from the fuselage and men were set to work with hacksaws cutting away any tubes that had been bent or dented. Fresh tubes were obtained and suitable lengths were sawn off, *the fresh lengths being inserted in place of the original tubes that had been removed, and welded into place.* The process sounds somewhat risky, and we frankly confess that until we had visited the testing department of the Fokker works we felt very skeptical about this method of effecting repairs. Mr. Stephan himself was, we believe, rather astonished when the repairs turned out to be successful, and from that day onwards he began to become converted and to see the real merits of the Fokker system of welded-steel tube construction, with which, since joining the firm, he has, of course, become more intimately acquainted and concerned.

In the laboratory of the Fokker works we were privileged to see a series of tests being carried out on samples of steel tube, partly mechanical and partly chemical tests, to ascertain the composition of the material, *i.e.*, carbon contents, etc., and partly tests to ascertain the strength of various welded joints.

With regard to the former it may be remembered that Mr. Fokker, in his lecture before the Institute of Aeronautical Engineers, gave the composition of the steels which he employs for his special form of construction, and that these are of a type that would not be accepted by the British Air Ministry,

being mainly more or less in the nature of ordinary mild steels. For the purposes of the special form of welded construction developed by Mr. Fokker, this grade of material has been found preferable to higher-grade materials which could not be welded conveniently, or, at any rate, with the same facility.

As regards the tests on welded steel joints, we witnessed both tensile and compression tests, and in no single instance while we were in the laboratory was the joint itself found to fail. In the compression tests the tube walls on each side of the welded joint "concertinæd," while in the tensile tests the tube broke on one side of the welded joint, but never in the joint itself. It would, therefore, appear that even for joints designed to work in tension the welded joint, which, by the way, was in the form of a perfectly plain straight-forward butt joined without overlapping, is perfectly satisfactory, provided an allowance is made for the decrease in the strength of the material itself, caused, of course, by the heating during the welding operation. One of the accompanying photographs shows such a test piece, and it will be seen that while the joint itself is perfectly sound, the tube on one side has contracted during elongation, while on the other side of the joint the tube itself broke. The actual amount by which the strength of a tube is decreased under the action of heat varies according to the composition of the steel, but, generally speaking, it is, we understand, round about 20 per cent. In other words, in stress calculations, instead of using the figure representing the strength of the untreated tube, about 80 per cent of that figure is taken. Other photographs show compression test pieces, and in no instance has the welded joint itself failed. Finally, one of the photographs shows a most complicated joint, or what would have been a most complicated joint with methods other than welding. In this joint no less than eight members meet, not counting the undercarriage struts which are hinged to the lower longeron by the hinges shown in the photograph. The bolt joint on the left is of the detachable engine mounting, which is now so designed in all Fokker makes so that it can be removed by undoing four bolts.

It is probably too much to expect, but we cannot help hoping that the British Air Ministry may be persuaded to abandon its present attitude toward welding. Not that we suggest that the British aircraft constructors should slavishly copy the Fokker system, but we do think that they should be permitted to develop methods of their own, somewhat along the same lines.—"Flight," July 16, 1925.

THE WING-ROTOR.

Considerable interest was created towards the end of last year by Herr Anton Flettner's application of aero-dynamical principles to the wind propulsion of ships, and his experiments with the rotor ship *Buckau*. The Magnus effect upon which its principle of propulsion depends was discussed in "The Engineer" of November 21, 1924, and a summary of Herr Flettner's paper before the German Institution of Naval Architects was given in the following issue. Throughout Herr Flettner's experiments a cylindrical rotor was employed.

An interesting development of the rotor principle has been made by Engineer-Captain Sigurd J. Savonius, of Helsingfors, Finland, who has invented and tested a new form of rotor which he terms the "Wing-Rotor." In contradistinction to the plain cylindrical rotor, which must be rotated by motor power and moved laterally if the Magnus effect is to be usefully employed, the wing-rotor is claimed by the inventor to produce by itself rotary motion as well as the Magnus pressure effect. From November last until June extensive trials of the wing-rotor have been made, and some of the results are set forth in the following. If an ordinary cylindrical rotor

be cut into two halves, which are then laterally displaced so that one outer and one inner edge is made to form a continuous wall, the S-shaped arrangement—illustrated in Figure 1—is obtained. That diagram shows the pressure distribution around the surface of the wings when the rotor is placed in an air stream. It will be seen that a region of positive pressure is formed near to the hollow face of the wing B, the turning effect of which pressure is, to some extent, annulled by the region of negative pressure or vacuum produced on the hollow face of the wing A. When a wing-rotor of this type is allowed to spin idly in the wind, its peripheral speed is found to be almost exactly equal to that of the wind speed, while if the two wings are

FIGURE 1.

FIGURE 2.

moved further apart, so as to make a space between them, the speed drops as the intervening space between the wings is increased. The obvious defects of such an arrangement are entirely overcome, so the inventor informs us, if the two halves of the wing-rotor are drawn apart to a distance equal to the length of a radius—as shown in Figure 2. With this wing grouping the air stream, which strikes the hollow surface of the wing B, is now deflected, so that it can exert a positive pressure on the surface of the wing A, instead of the negative pressure acting at that point, shown in arrangement, Figure 1. The effect of this seemingly small alteration is, we are informed, a very marked one. With it an increase of the peripheral speed of the wing-rotor, up to 1.7 times the wind speed, was noted, while the turning moment was shown to be three times that of a similar wing-rotor without a center opening—see Figure 1—and five times that of a rotor in which the two wings were separated by some distance. Two half-cylindrical wings were found to give the best overall efficiency. Further comparative tests between a wing-rotor and a cylindrical rotor showed, we understand, that the Magnus effect was the same for rotors of equal length and surface area when both were rotated at the same speed. As in the case of the cylindrical rotor, the efficiency is improved by fitting end plates to the wing-rotor, so that the air does not spill over the ends. It is claimed that with Mr. Savonius's wing-rotor the diameter of end plate required is only from one-quarter to one-third larger than the wing spread, or smaller than with the plain rotor.

The trials which have been made indicate that the wing-rotor is specially suitable for wind-motor work, for, compared with the windmill, it has fewer working parts, less frictional loss, and requires no special vane apparatus to keep it to the wind. Only two simple ball bearings are required, which

can be so arranged that lubrication can be effected from the ground. The speed of rotation of a wing-rotor can be fixed within wide limits by altering the ratio of diameter to length. Thus the speed of a short and broad wing-rotor is less than that of a tall and narrow one. When fixed to the top of a mast—as shown in Figure 3—the proportion of width to height may be 1:1 to 1:2, while if mounted on a low basement on the ground a length of three to five times the height may be employed with advantage. A simple brake is employed to keep the wing-rotor stationery when it is not required. When allowed to come to rest the wings automatically take up a position of least resistance, which is such that the common diameter of the two wings stands at angle of 25 degrees from the direction of the wind.

The following example of a wing-rotor arranged for water pumping may be given. The wing-rotor to be tested was fastened on the top of a bollard on a light pier, and it had the following principal dimensions:—Height between the end plates, 196 centimeters; wing spread, 96 centimeters; diameter of end plates, 120 centimeters; area, 1.88 square meters. The wings were constructed of 0.75 millimeter sheet iron, with the edges wired with 10 millimeter wire. Four 3 millimeter wire stays were used on each wing, and the end plates were made from 12 millimeter plywood. The rotor itself was supported on a 2-inch steel tube, to which it was attached by concave sheet metal covers 1 millimeter thick. There were two ball bearings, the lower of which was designed to take the weight of the wing-rotor, 50 kilos. At the lower end of the rotor was fixed a circular plate with an adjustable arm to vary the length of stroke of a reciprocating pump, which had a piston diameter of 34 millimeters. With the longest pump stroke the amount of water delivered was 50 cubic centimeters per stroke. The water was pumped along a 366-foot ¾-inch main to a tank 50 feet above the suction level. At a wind speed of three meters per second, or about 6¾ miles per hour, the pump gave 36 strokes per minute and delivered 100 liters per hour. At four meters per second, or 9 miles per hour, wind speed, the output was doubled, and it was again doubled when the wind speed rose 6 meters per second, say, 13½ miles per hour. These figures, it is claimed, compare very favorably with the performance of a 6-foot steel windmill of modern design, and the output may be increased by using a larger wing-rotor. The inventor has also produced a self-regulating wing-rotor in which the wings are pivoted on tappets arranged on the end plates, and the edges of the wings connected by tension rods again pivoted at the edge of the wing. Two pairs of springs are fitted which hold the wings apart, and a chain attached to one of the wings runs down the hollow tube support to a hand windlass. Should the speed of the wing-rotor rise the centrifugal force acting on the wings pulls them round, so that less of the wing surface is exposed to the wind, whilst if the speed again falls the springs bring back the wings into the original position. The rotor can be brought to rest by the hand windlass, which brings the two wings into neutral position.

The inventor has also tested the propulsive force of the wing-rotor in a small boat about 16 feet long and 6 feet beam, in which two wing-rotors, with a width of 67 centimeters and a height of 280 centimeters were fixed. The end plates were 88 centimeters in diameter. In a breeze of 7 meters per second, or 15 miles per hour, the boat reached a speed of 5 knots and at a wind speed of 10 meters per second 6 knots was averaged, equal to a propelling force of 3 to 4 horsepower. The experimental work we have referred to has, it appears, shown that the wing-rotor offers various means of utilizing the wind power in a novel and simple manner. The apparatus is, we understand, protected by world patents, and further experimental work with regard to its application is to be carried out by the inventor.—"The Engineer," Aug. 21, 1925.

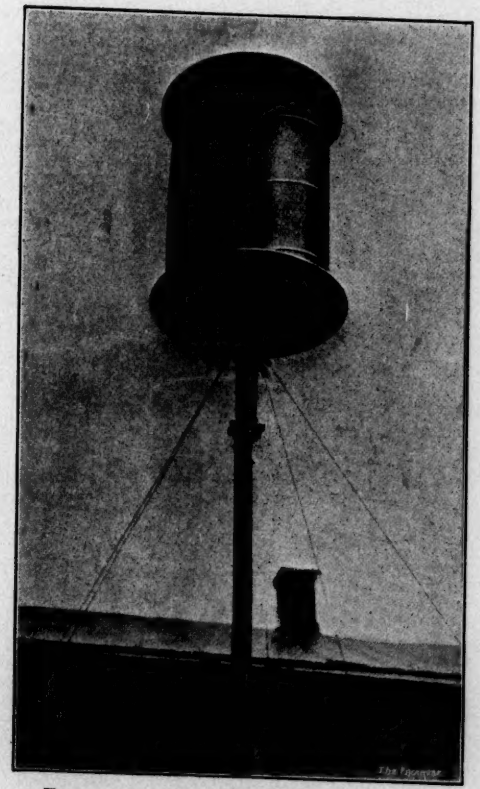


FIGURE 3.—A WING-ROTOR PUMPING UNIT.

SEE WITH YOUR RADIO.

By W. B. Arvin.

The last week in June, seven men stood in a laboratory in Washington, D. C., watching the arms of a miniature windmill revolving on a small screen of white blotting paper. The real windmill was five miles away in Anacostia. The picture on the laboratory screen was being transmitted by radio through the intervening space.

It was the first time in history that real television had been demonstrated or motion pictures had been transmitted by radio—the little moving windmill picture.

The men present at the demonstration were Mr. D. MacFarlan Moore, inventor of the lamp upon which the receiver depends; Mr. C. Francis Jenkins, inventor of the prismatic disc used in the transmitter; representatives of the Navy Department; Mr. Burgess, Director of the United States Bureau of Standards, and Judge Taylor, advocate of the Department of Commerce, who presides over radio under Secretary Hoover.

After the demonstration, everyone was unanimous in the opinion that this long-sought-for goal is at last actually in sight. In a more or less perfect form it will be a common thing within a year. The main difficulties in the problem have been successfully worked out.

No few pages have been written on this subject. Indeed, it is a barren month which does not find the radio press carrying some sort of scheme for the transmission of pictures by radio. But the point with most of them is that they are usually very plausible on paper but hold little promise of practicability when reduced to practice. The present scheme is practical. It works. This is the real point.

Probably the most outstanding characteristic about the whole scheme is its simplicity. Great inventions are usually simple, however. The success of their makers is primarily a matter of removing the complexities. With the Jenkins and Moore machine, simplicity is noticeable from the first.

As to the technical description, it can be made plain enough for the man in the street to understand.

The first of the two prerequisites of any picture transmitting apparatus is a photo-electric cell which will take the blacks and whites of the photograph or picture to be transmitted and turn them into electric current in direct proportion to their shading. The whites must transmit a heavy current when it is focussed on the cell, while the intermediate grays and the blacks must transmit little or no current in accordance with their approach to dead black.

With electric energy in direct proportion to the shading of the picture, the transmission of this energy to a distant point is a comparatively simple matter. We may use radio or fall back upon the land-line.

Then the second point of importance is a lighting device at the receiver which will take the currents after they have been amplified, and turn them again into light and shadow in exact step with the transmitter and in exact synchronism with the shade changes in the picture at the transmitter.

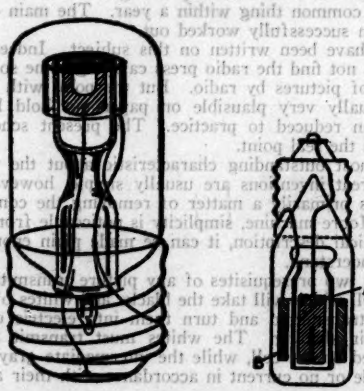
Practically, these two problems have been solved very well. The light at the receiver is one developed by Mr. Moore. It depends upon the glow about the negative pole of a lamp in an inert gas, such as neon. The characteristics of this little tube are such as to make it excellent for television—or telerama, as Mr. Moore chooses to call it.

Tests in the laboratory have shown that the lamp may be lighted or extinguished at the rate of more than a hundred thousand times a second. The superiority of the lamp in this direction may be appreciated when one

calls to mind that with the ordinary hot filament type of light, more than one-tenth of a second is often required for the hot filament to radiate sufficiently for the glow to cease. Its second point of superiority is its ability to follow exactly in illumination the current fed to it.

In one stroke, this little lamp, which is shown in an illustration, eliminates two very complicated and bulky pieces of apparatus which have heretofore been incorporated in television receivers. Previously, in order to get comparative light and dark, it has been necessary to install and use some sort of magnetic shutter to govern the amount of light passed to the screen. The Moore lamp remains in exact step and does away with the quantitative shutter.

At the transmitter, the photo-electric cell which turns the light and darkness of the photo or scene into electrical energy is a commercial product. It seems that there is a great deal of misinformation afloat concerning the effectiveness of present-day photo-electric cells. Two large manufacturers have in the open market at the present time two designs of these instruments which will fulfill almost any conditions which might be imposed upon them. One of these cells is employed in the Moore-Jenkins apparatus.



TWO TYPES OF THE MOORE TELEVISION LIGHT.

With a capable method of converting the light into electric energy and back again into light, the only other important consideration for the operation of the machine, is that some method be employed for breaking the picture up into small signals. This simply means that the picture must be transmitted in dots, or that it must be broken up in some fashion so that a large number of impulses are employed to transmit the picture. In the last analysis, it simply amounts to telegraphing at high speed.

This part of the process may be likened to the use of half-tones in the printing of photographs. Everyone knows that a half-tone consists of innumerable dots. Yet when the printing process is complete, the reader does not see the dots, he simply views the composite effect of them. It is

the same with the transmission of pictures. The great number of dots, differing in their intensity, tend, when put together properly, to give the complete effect of a picture.

This breaking-up process in the Moore-Jenkins machine is accomplished so simply that it is almost absurd. With the aid of the prismatic disc spoken of before and a second ring containing a number of lenses, the process is done with the least possible complication and with the greatest clarity and ease. The lenses—there are forty-eight of them—are focussed so that they draw a fine line across the face of the picture. Each of them would follow the same path were it not for the prismatic disc. The forty-eight of them cover a picture about four inches long. That is, each one covers approximately one-twelfth of an inch of the picture.

The prismatic disc moves these lines struck out by the lenses from the top to the bottom of the picture, giving the complete break-up process.—"Radio News," Sept., 1925.

ON RECENT ADVANCES IN WIRELESS PROPAGATION BOTH IN THEORY AND IN PRACTICE.*

By A. S. Eve, D.Sc., F.R.S.

Director, Department of Physics, McGill University; Associate Editor.

ADVANCES IN PRACTICE.

The use by Marconi of vertical reflecting wires, arranged in a parabola with the transmitting aerial in the focus line, has resulted in remarkable practical applications. In the case of a plain aerial, if 10,000 units are required to reach a given distance, then with a two-wave aperture parabolic reflector only 25 units are required, whereas with an eight-wave aperture only 1.56 units are required. Hence with power of 18 kw. on the oscillator valves and with a wave-length of 92 meters, large distances, such as from Cornwall to Buenos Aires, 5820 nautical miles, have been reached, and on this wave-length daylight signals have been reliable, and night signals stronger than were expected.

Another method which is now being employed is that of a network in a vertical plane resembling a gigantic tennis-net 3000 feet long and 300 feet high. L-shaped conductors on this net are insulated and tuned to the desired short wave-length. Behind this, at a distance of a quarter of a wave-length, is another plane network used as a reflector. By this method, instead of using lofty towers 820 feet high with 250 kw. on the aerial, involving the use of expensive insulators, it is sufficient to put about 25 kw. on the network in order to obtain equally good results, and the engineering problem is also much easier.

It is somewhat remarkable that experiments indicate that waves of a hundred meters give bad results during the daylight hours, whereas waves of thirty meters give good results by day, but bad by night. With an intermediate wave-length of about 45 meters, results are equally good both by day and by night, so that this wave-length is likely to be employed for long-distance signalling, as from Canada to Australia. It is noteworthy, too, that Macmillan in Alaska is to communicate next July with Schnell on the *Seattle*, U. S. Navy, in the antipodes, off Australia, with a wave-length selected as 20 meters. In fact, the day of the short wave-length has arrived!

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* Communicated by the Author.

Experiments at Hendon, employing 700 watts with 4000 volts on the transmitters, have rendered good speech at 97 miles with a wave-length of 15 meters. Atmospherics are not troublesome, unless a thunderstorm is actually in sight, but trouble has arisen from the magnetos of automobiles emitting 15-meter waves which produce clicks in the receivers. Waves as short as 4 meters have also been tried.

At Inchkeith, in the Firth of Forth, two rotating parabolic reflectors have been mounted back to back on a vertical column with an aperture of 13 meters, which is about double the wave-length 6.3 meters. These reflectors revolve and give Morse signals for various compass bearings, so that a ship at sea is able to pick up its bearing from the land. It is probable that this method of signalling, by which a ship can obtain its own position from a revolving radiator on land, will be extended.

The amateurs have certainly developed with efficiency any bands of frequency assigned to them. It is not possible for them to use parabolic or plane reflectors, and so they have spread their radiations in all directions, frequently attaining success with remarkably low powers, for example, from England, to their antipodes in New Zealand with a quarter of a kilowatt, from Montreal to British Columbia, with 15 watts input and 5 watts output, and similarly from San Francisco to New Zealand. Best of all from Halifax to England with 2 watts!

Mr. J. Miller, of Montreal, signals to New Zealand with 100 watts and to England with anything from 30 to 110 watts.

He states that with the sun shining on his aerial he gets good results with 5-meter or 20-meter waves, but that the signals are poor in cloudy weather and fail entirely at night when he perforce lowers his frequency. There does not appear to be any known reason for this remarkable effect, which he states is a common enough experience. He states that 20-meter waves give him the best results by day. Miller states that with short waves, say 5 meters, there is frequently no reception within about 500 miles, but results are obtained at greater distances, as though the waves first mounted upwards and were then reflected or bent downwards.* Appleton makes similar statements.

Pittsburgh broadcasts both on 62 meters and on 309 meters, so that comparison can be readily made between these waves. The shorter wave (62 m.) conveys speech admirably at short distances, but the results are described as "mushy" at greater distances, at Montreal, for instance. Indeed, some amateurs believe that the higher audio tones travel faster on the carrier wave than do the lower audio tones, a difficult matter to explain, if correct.

For transmission amateurs do not use the ground but a "counterpoise" or series of wires beneath the main aerial, the two sets being 60 and 20 feet above the ground for example. In other cases the aerial is vertical, ungrounded, with a coil in the middle coupled inductively or by capacitance with the transmission set. The skill and perseverance of amateurs is worthy of more recognition, and their results have great scientific interest.

ADVANCES IN THEORY.

The old question as to the method by which wireless waves travel round the earth has had much life given to it by a valuable discussion and theory due to Sir Joseph Larmor, published in the "Philosophical Magazine" for December, 1924.

Again, in the "Proceedings of the Physical Society of London" (February 15, 1925) there is a full summary of a joint meeting of the Physical Society of London and the Royal Meteorological Society. This summary

* I learn that rays from Pittsburgh to Australia pass Fiji, but cannot be detected there. (See also the papers of Reinartz in *Q. S. T.*)

should be read, because it throws much light on so many interesting and important topics. The speakers included Eccles, Chree, Appleton, C. T. R. Wilson, Simpson, Chapman and others.

A brief summary will be here given indicating the modern viewpoint.

In June, 1912, Eccles* first gave a theory of the Heaviside layer of ionized gas, highly conducting and supposed to be capable of reflecting electromagnetic waves in the higher atmosphere, while in the middle atmosphere the ionization is assumed to be sufficient to bend the rays round the earth. Eccles attributed the bending to the small oscillations given by the radio waves to molecular ions.

Today Larmor assigns the bending to light ions, such as hydrogen ions, or to electrons, or to both.

Fifty miles (80 km.) up in the atmosphere the free path of molecules between collisions is 2 cm., while the number of molecules per cm.^{3} is 10^{15}, as compared with 3×10^{19} near the earth.

Larmor shows that during the time when the light ions or electrons are moving quite unimpeded from one collision to the next, there is time for the wireless waves to set them into vibration with a not insignificant amplitude (which varies with the square of the wave-length). These oscillations constitute a current of a character quite similar to a Maxwellian displacement current. The true dielectric constant is decreased, and the velocity of the wireless wave is correspondingly increased.

Hence with correct adjustment of the transmitter the rays can be directed round the earth to distances such as 13,000 miles, travelling in their journey between the earth and the conducting layer about 50 miles above it. At the best an aerial can draw on this radiation over an area of the order of the square of the wave-length employed. There is sufficient energy for all receivers.

The dielectric constant, K, becomes the effective K′ where

$$K' = K\left(1 - \frac{N^2 e^2 \lambda^2}{\pi m}\right)$$

Here N is the number of electrons or light ions per cm.^{3}, e is the charge and m the mass of any one of them, while λ is as usual the wave-length.

Larmor shows that even *one* freely moving electron per cubic centimeter at the upper limit would suffice for the requisite bending round the earth, especially as he points out that free electrons can and do frequently pass through atoms without collision with the constituent parts of the atoms.

On the other hand, it appears that we must abandon the Heaviside layer as a reflector! Theory indicates that it would absorb and scatter and hence destroy wireless waves, rather than reflect them. Yet most practical radio experts find strong indications of reflected waves.†

Appleton suggests that the rays are bent rather than reflected by the Heaviside layer, so that we have a mirage rather than a reflector influence.

Chapman points out that he finds from magnetic results that the conductance of the upper layers is equivalent to a meter thickness of copper over and around the earth. This layer is penetrated by short wave radiation such as light, but it would presumably not be penetrable by radio waves.

If radio waves go half around the earth with the velocity of light (c) then the upper part of the waves have a longer path to travel, and to keep up with the lower parts they must necessarily move faster than light. This at first appears disconcerting to the relativist! But the difficulty is capable of explanation.

* "Proc. Roy. Soc.," p. 86.

† In a later letter to "Nature," Larmor points out that with fairly abrupt transition to a well-ionized layer there may be reflection of wireless waves analogous to the reflection of light from metals.

ATMOSPHERICS.

C. E. P. Brooks has collected valuable statistics on thunderstorms. A single station reports on an average 16 thunderstorms a year over an area of about 200 square miles, so that the whole earth has about sixteen million thunderstorms a year, or 44,000 a day. If on the average they last for an hour, then there are always 1800 thunderstorms in being, and allowing 200 flashes per hour per thunderstorm, we find 360,000 flashes an hour, or 100 flashes a second over the whole earth.

These will be sufficiently numerous then to disturb the most enthusiastic radio fan! For example, around London for a radius of 1000 km. there will be about one flash a minute in winter and about a dozen a minute in summer.

South Africa and South America appear to be the places where the greatest number of thunderstorms occur.

It seems quite certain that lightning flashes do cause atmospherics, but other causes may exist. For example, C. T. R. Wilson points out that there may be silent but abrupt discharges between the Heaviside layer and the upper regions of a thundercloud.

Appleton and others have taken observations on atmospherics with the Western Electric cathode-ray oscillograph. The disturbances are of four types indicated by Figures 1-4.

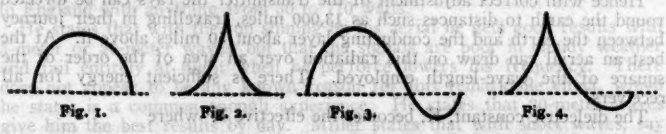


Fig. 1. Fig. 2. Fig. 3. Fig. 4.

The periods of these oscillations are measurable in millionths of a second, or microseconds (μs), and occasionally in thousandths of a second, or milliseconds (ms.). The electric forces are not large, the main type being of the order of 1 volt for 8 meters, but at Khartoum as much as 250 volts has been received on the aerial, or 4.2 volts/meter.

An average lightning flash consists of from 3 to 6 discharges lasting each about 3 milliseconds with quiet intervals of a similar period intervening.

EXTINCTION OF SIGNALS.

Austen has accounted for the weakening of wireless waves with distance by an extinction factor $e^{\frac{-\alpha r}{\sqrt{\lambda}}}$ involving the distance of propagation r, and (it will be noted) the square root of the wave-length. The constant α has a value of .28 for dry land or sand, and .0025 for water where r and λ are measured in kilometers. It is important to have a good conducting ground around the transmitting aerial so as to launch the radiation with an upright wave front.

Inasmuch as signals travel with less loss between flying aeroplanes than between ground stations, it appears that it may be desirable to launch waves tilted upwards from the ground.

ON WANDERING WAVES.

In addition to the Larmor bending effects due to light ions, it seems to have been proved that the earth's magnetic field, of the order of one-half a gauss, provided that ions or electrons are present, institutes a change of

direction which will be least in the magnetic meridian and greatest for propagation at right angles to that direction.

Theory indicates that the plane of polarization may be rotated to an amount dependent on the electron density, the magnetic intensity and the frequency.

Hence we import into radio theory all the familiar peculiarities of double refracting crystals in physical optics, with rotation of the plane of polarization, an ordinary and an extraordinary ray, etc. At a critical frequency long waves may bend down, while short waves bend up. A may be able to signal to B, but B may not be able to signal to A!

We find here ample reasons for the violent errors in direction finding about the hour of sunset which are so troublesome in practice, and which were so obscure in theory. Nichols and Schelleng, of the Bell Telephone Company, have called attention to these explanations in an article to "Nature,"* but much material had already been published in the "Physical Society of London Report" (February 13, 1925).†

It is interesting to find the theories of atmospheric electricity, earth's magnetism, and wireless waves all intermingled so that an advance in any one of them is likely to throw light on the other two, and this fact suggests the desirability of bringing atmospheric electricity within the scope of the meteorological offices of all nations.

In conclusion, I point out with delight the truly international and co-operative character of the brotherhood of radio investigators. When this spirit pervades further the people, the politicians and the governments of the nations of the world, we may begin to think and work internationally, and so bring about an age of thoughtful adjustment of difficulties instead of blind prejudice leading to hatred, war, waste and the possible destruction of such civilization as man has so far managed to achieve.—"Journal of the Franklin Institute," Sept., 1925.

PARSONS GEARED OIL ENGINE SYSTEM.

The number of British oil engine builders is constantly increasing. In which direction the oil engine for ship propulsion shall eventually develop is at present impossible to predict, although it is apparent that the opposed piston type in its recently simplified form becomes a strong competitor to the standard double-acting engine of either four or two-cycle type. But even should this type for some reason cease to exist, British manufacturers are still far away from the type of oil engine that might become standard, as the modern steam engine is already, since it is improbable that the various licenses of one given type would develop their large sizes, especially the double-acting types, along the same lines.

That the oil engine of today becomes a strong competitor even to the steam turbine is evident from the fact that such important firms as Parsons Marine Steam Turbine Company, Ltd. (Turbinia Works, Wallsend-on-Tyne) enter the field of building oil engines for ship propulsion. This seems to add weight to the prophecy of one notable authority that in 5 years no more orders for steamships will be placed.

The Parsons oil engine unit is exceedingly interesting in a number of ways. It is easily comprehensible that the valuable experience with the Parsons' geared drives has been made use of. There is a division of views, however, on the relative advantages of using high speed oil engines in place of the slow speed direct connected units now mostly used in the merchant

* March 7, 1925, pp. 334-335.

† They give a clear and forceful statement (pp. 215-234) in the "Bell System Tech. Jour." for April, 1925.

marine. The advocates of high speed engines consider it better to have smaller cylinders but in larger numbers, instead of few cylinders with very large dimensions. For ship propulsion especially does the high-speed engine offer decided advantages. Main requirements of a good marine oil engine are: light weight and small space, simple, reliable and easily accessible construction, low purchase price, low cost of operation and upkeep. The larger number of cylinders should not necessarily mean higher cost, engines of equal horsepower with different number cylinders can be easily built so that the one with the larger number of cylinders will weigh less and hence will entail even smaller cost of upkeep.

The Parsons' geared oil engine system employs a number of high-speed four-cycle engines of enclosed type with pressure lubrication. The output of each engine is transmitted by means of a single reduction gear to the propeller shaft. An elastic friction coupling built into the flywheel together with an easily operated friction clutch permit the withdrawal of each individual engine from the propeller drive. Two or four engines are used to drive the propeller shaft, the pinions connected to the engine shafts engage in the customary manner the big gear rigidly connected with the propeller shaft. The main data for both cases is given in the following table:

Type of engines	2 engines 4-cycle, high speed, crosshead, closed crankcase.	4 engines 4-cycle, high speed, trunk piston, closed crankcase.
Horsepower of each engine	600	300
No. of cylinders per engine	6	6
Total horsepower to the propeller	1200	1200
Total weight including gear drive and propellers, tons	261	226
Length of engine room	32 feet (+5 feet operating room length)	26 feet, 9 inches (+5 feet operating room length)
R. P. M. of engine	300	450
R. P. M. of propeller	75	75

In this manner both oil engines and propellers work at their most efficient speed (R.P.M.). Smaller engine parts are more easily handled and when any one of the engines is in need of repair it is disconnected without disturbing the operation of the others.

Besides, for any desired speed of the boat the load may be distributed among such number of engines, by connecting or disconnecting without any difficulty, that they will operate at their maximum efficiency.

Another very notable departure from the standard oil engine designs is the extensive system to reclaim exhaust heat. Cylinder jackets, vertical exhaust heated regenerators and an oil-burning boiler constitute one closed system. Each engine has its own regenerator. Before starting, steam is raised in the oil-burning boiler. Hot water is pumped through the jacket spaces, warms up the cylinders and makes starting easier and more reliable, thus permitting the reduction of compression pressure in the working cylinder. Besides, the heat stresses are thereby reduced and the reliability of the installation increased, the higher cylinder temperatures increase the thermal efficiency thus compensating for a slight reduction in efficiency due to higher engine speeds and smaller compression pressures. The cylinder jackets are designed for pressures up to 250 pounds per square inch and their preheating permits the use of steam for starting, which is admitted in the regular manner through starting valves in the cylinder head.

Due to high temperature of the cylinder, starting is very easy and quick with small consumption of steam. The experimental engine equipped with this kind of starting arrangement performed in all respects highly satisfactorily and demonstrated its superiority to the ordinary type of oil engine starting with compressed air. Reversing and all maneuvering is also done by steam and offer neither practical nor technological difficulties. When the engine is in operation the waste heat from circulating water and exhaust gases is used to raise steam necessary to drive auxiliaries, it is quite sufficient for that purpose and thus no extra expense for driving the auxiliaries is necessary. Steering engine, capstan, circulating water, ballast, bilge and other pumps are all driven by steam generated by reclaimed waste heat. Evaporator, surplus steam condenser, auxiliary steam-driven air compressor and steam-driven generator (the other end of it is coupled to a small hot bulb engine as a reserve) as well as lubricating and fuel oil pumps complete the number of auxiliaries. In one of the proposed installations the lubricating oil and circulating water pumps are driven by the gear drive.

There is no doubt that such a combination of steam and oil engine drive as used here and in the Still engine, though in quite a different manner there, can work very efficiently. It would be interesting to compare the Parsons and Still engines in this respect. It is true that so far there is no order placed for the installation of one of the Parsons' geared oil systems on board ship. The fact, however, that it offers a possibility to reduce the cost of ownership leads one to presume that it will soon be given an opportunity to be tried out in actual service on the high seas.—"Schiffbau," 9 September, 1925, transl. by E. C. Magdeburger.

THE WHALEY CONSTANT PRESSURE OIL ENGINE.*

The Sun Shipbuilding and Dry Dock Company of Chester, Pa., has completed the building of the new Whaley oil engine, which was invented by Mr. William B. Smith Whaley and designed by Charles A. Muller of The Whaley Engine Patents, Inc. This engine was privately shown to stockholders and interested engineers, at which time it was operating on air only for the shop tests.

Ninth of a series built during the past several years for the purpose of perfecting a new power cycle, this Whaley engine is described as a single-acting four-cylinder two-cycle machine designed to develop 750 horsepower. It is a vertical reversing engine, conventional in design though following the best steam engine practice. It is distinguished from all other heavy oil engines in this country and abroad by the fact that the cylinder clearance, at the time of the fuel injection, is opened by a piston valve to communication with a static receiver having many times the capacity of the cylinder clearance. This presents a condition exactly the opposite to that in all present internal combustion engines, where the basis of operation is on a trapped charge. In the Whaley engine, the piston valve to the static receiver is open as long as the fuel is being injected.

The Whaley engine has the usual form of port scavenging supplemented by high pressure scavenging and super-charging through the top of the cylinders. The last two operations are carried out with air at a pressure of about 45 pounds per square inch. The air pumps for the super-scavenging and super-charging operation are contained in the lower part of the power cylinders. There is a supplemental scavenging air pump formed in the upper part of the frame in which operates a piston placed on top of the crosshead.

* This article is contributed by Parish & Tewksbury, Inc., 17 East 42nd Street, New York.

Two cylinders are cast in block with a valve chest between. The air from the pumps just described is used either in direct line or is transferred through the ports in the valve chest to the opposite cylinder.

CONTROL OF THE AIR.

The air is controlled by piston valves, one working inside the other and both being actuated by links similar to those used on locomotives and reversing marine steam engines. By linking up, the opening of the piston valves is made to synchronize to the period of fuel injection in such a way that, for practical purposes, combustion takes place during the entire time that the valve is open. The practical effect in the operation of the engine will be that fuel will be supplied to the point of cut-off of the piston valve, which will be open for the power necessity of the engine. The result-diagram will, therefore, be that of the steam engine so far as the admission and point of cut-off are concerned. The main purpose of the clearance reservoir is to maintain the maximum working pressure desired, which, in the case of the present engine, is 350 pounds per square inch. As this is the normal compression pressure of each working cylinder, it is also the normal pressure for the receiver or reservoir.

Under normal running conditions, no variation in pressure is expected no matter what the conditions of loading and fuel injection may be. In floating the receiver on the compression space, there will be but little transfer of air, heat or pressure through the valves. The receiver serves as an absorber in case of any rise in pressure in the cylinder through any cause whatever. By an ingenious arrangement of the valves, the compression space port is made to serve successively for the introduction of the super-scavenging, and super-charging and to establish communication with the reservoir or receiver during the combustion period.

NOVEL FUEL PUMPS AND VALVES.

A novel arrangement of fuel pumps and fuel admission valves is used. The fuel pump is placed on top of the valve chest and is driven directly from the valve stem. The pumps deliver the charge of oil for each stroke directly into each individual fuel valve, there being no piping whatever employed. The pressures will be between 4000 and 8000 pounds per square inch. The action of the pumps synchronizes with the action of the valves, both being operated from the same valve stem, thereby assuring the maintenance of relation between fuel injection and duration of the valve opening to cut-off. Because of this construction, there practically will be no rise in pressure in the cylinders or in the system beyond that due to compression.

FUNDAMENTAL ACTION.

The fundamental action of the Whaley engine is entirely different from that of any of the present oil or Diesel engines. All other present-day engines operate with a trapped charge of air which is confined in closed compression space into which the fuel is injected, and there is always a rise in pressure after combustion takes place. In the Whaley engine, even early timing of the fuel admission will cause no appreciable pressure rise above the capacity of the pressure valves into the receiver.

Maintaining the maximum pressure to a definite figure allows the carrying out of a program for the design of relatively light weight oil engines. It is expected that the Whaley engine will weigh less than 100 pounds per horse-power, a figure which represents a substantial improvement over current practice. Also, due to the maximum-pressure principle, the Whaley engines, for all sizes of power, can be built at approximately the same weight per

horsepower as is the case with the steam engines. This means that the weight of less than 100 pounds per horsepower will apply to Whaley engines of all sizes. This feature will make the development of the Whaley engine much more valuable as an economic factor, the larger the units are built. Especially so as with the present Diesel and heavy oil engine weights increase progressively as the power that is to be delivered increases.

PRESENT STATUS.

The Whaley engine at the Sun Shipbuilding and Dry Dock Company's plant is now undergoing its shop tests, and it is expected that the trials will be completed in time to present all technical details of the engine and the results of performance before the winter meetings of the national engineering societies.

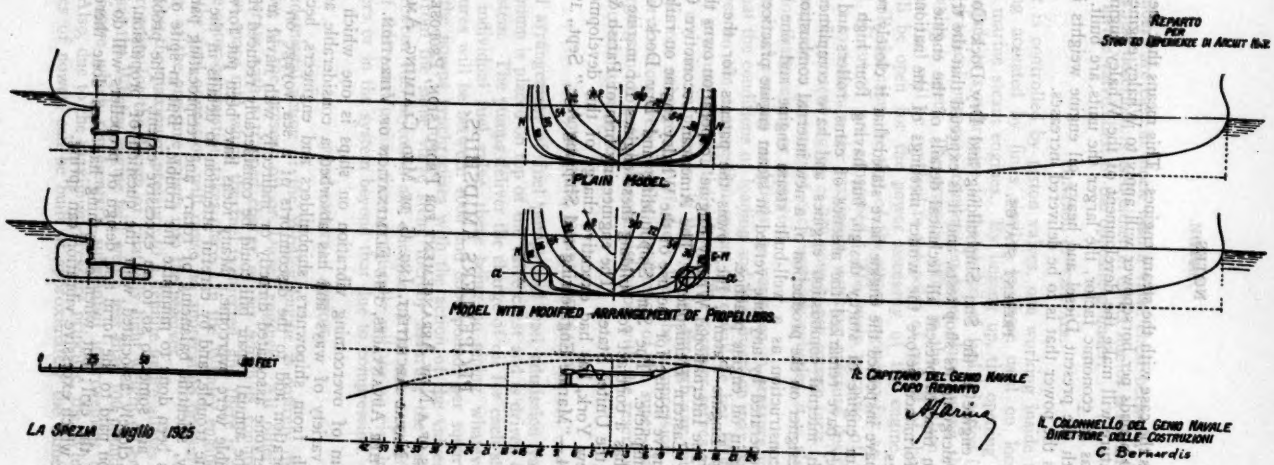
Those who have inspected the engine have stated that it closely resembles a vertical steam engine of sturdy design, but having four high pressure cylinders. They have remarked the absence of cams, rollers and gearing associated with internal combustion engines and have complimented the inventor and designer on the production of a new internal combustion engine as simple in construction as a well-built steam engine and one that can apparently be operated by any one versed in steam engine practice without special training in oil engine practice.

The Whaley Engine Patents, Inc., owns the patents for the Western Hemisphere. The International Whaley Engine Corporation owns the patent rights for the Eastern Hemisphere. The American Locomotive Company owns the exclusive license for the use of the Whaley engine on rails for the Western Hemisphere. The Sun Shipbuilding and Dry Dock Company, Chester, Pa., has a contract for building these engines for marine and stationary use in the United States. The engineering firm of Parish & Tewksbury, Inc., New York, has had executive charge of this development since February, 1924.—"Marine Engineering and Shipping Age," Sept., 1925.

PROPELLERS AMIDSHIPS.

FIRST DETAILS OF A NEW ARRANGEMENT FOR PROPULSION, PROPOSED BY THE ITALIAN NAVAL ARCHITECT, ING. G. DE MEO, CLAIMING, AMONG OTHER ADVANTAGES, THE ELIMINATION OF VIBRATION.

The problem of overcoming vibration on ships is one which has been tackled in a variety of ways and has absorbed a considerable amount of attention, both from shipowners, shipbuilders and engineers, because, not only does vibration add to the discomforts of a sea voyage on passenger ships, but everyone associated directly or indirectly with naval architecture knows how the annual repair bill could be considerably reduced if only the vibration trouble were overcome. Many ideas have been put forward as a solution of the trouble, and by careful attention to details in the design of the machinery, scientific balancing of rotary and reciprocating parts and so on, much has been done to minimize the trouble. But in spite of this, all ships vibrate, and some do so to an excessive extent. The problem is, of course, very closely associated with the question of propulsion, and here again attention paid to the form and design of propellers will do something to counteract the evil; but when everything has been done there are still sources from which excessive vibration can spring.



Lines of the Model with which Experiments were carried out at the Italian Naval Experimental Tank, Spezia.

An interesting proposal has recently been put forward in this connection by Ing. G. de Meo, who has conceived the idea of placing the propellers nearly amidships. The idea of driving a ship with propellers amidships is not, of course, entirely new, but Mr. de Meo has considered the problem in an entirely different light from that approached by some other experimenters, and it appears to us that his suggestion has reached a stage when it should be regarded as a scientific contribution to the many theories and propositions which have hitherto been put forward. Mr. de Meo is well known to a large number of our readers as a naval architect and engineer at Genoa, and he was largely instrumental in the establishment of the new Registro Nazionale Italiano, embodying an international agreement with classification societies of other countries in order to carry out registration and survey work with technical uniformity. Mr. de Meo, in the course of his practice, has been impressed by the amount of damage he has witnessed as the direct consequence of vibration, mainly due, he believes, to the rotation of the screw propellers, especially in ships of a limited coefficient of fineness where the action of the stream lines of water striking the propeller blades in their motion under the submerged part of the hull creates a great disturbance of the water. The evil is, of course, increased by the heavy pitching of the vessel in rough seas, and leads to bad steering and other defects. Proceeding with this argument, Mr. de Meo considers that when these vibrations are increased by the fast revolution of the propellers they produce evil effects on the bearings of the lines of shafting, besides causing special strains along the whole length of the vessel, and damage to the elastic structure of the hull and machinery, especially in modern high-speed liners and warships. Such vibrations cause great discomfort to the passengers, especially those in second-class cabins on modern ships—these cabins being generally located on the extreme after part of the vessels. It has to be remembered that modern ship types or systems of propelling machinery involve the adoption of screw propellers of a small diameter, necessarily increasing the number of revolutions of propeller shafts. After a series of experiments carried out at the Froude Experimental Tank of the Royal Italian Navy at Spezia, Italy, Mr. de Meo has finally attained what he believes to be a practical solution of the problem.

CENTRAL PROPULSION.

The basic idea is the central propulsion of the vessel, leading not only to the avoidance of vibrations, but to many other technical, economical and commercial advantages. Propellers of a new design have been developed. These are placed in special recesses formed in the sides of the vessel, and in direct connection with the geared turbines or other system of propulsion. They are so placed in the central portion of the submerged parts of the hull and at such a depth that they cannot emerge from the water level even during the maximum rolling of the ship. From data which we have seen, it appears that the experiments carried out at the Froude Tank at Spezia show that a much better utilization of power developed is obtained by the adoption of the new system of propulsion, and this is readily understood when it is realized that all appendages and V brackets on the submerged parts of the hull are dispensed with.

In a statement which Mr. de Meo has prepared in connection with his invention are the following claims, which we reproduce in their entirety:—

(1) That the elimination of vibration in the vessel aft of amidships adds enormously to the comfort of the passengers.

(2) That the present screw propellers at the stern of the ship are dispensed with, and as the propelling power is now placed amidships the usual long lines of shafting, with their bearings, tunnels, tail-shafts and acces-

sories, are done away with as no longer necessary. The saving so effected in efficiency of transmission of power, in weight, cost and space for cargo purposes is a very valuable feature of the new system.

(3) That all fractures in the tail-shaftings and intermediate shaftings, as well as the heating of white metal in the bearings and in the other parts of the machinery, due to vibrations, are completely avoided by the adoption of the new system, thereby adding to the life of the machinery. More particularly is this the case when vessels are "working" and pitching in rough seas.

(4) That the revolutions of the main engines are maintained at a practically constant rate, even when the ship is rolling and pitching heavily, and this, too, with any class of machinery and any type of ship.

(5) That by means of a new and original type of propeller it is possible with two propellers of the new type placed amidships to replace the three or four propellers placed at present at the stern of modern large liners or war vessels.

(6) That a great deal more protection is afforded to the propellers of merchant vessels and any type of naval vessel by placing them amidships, as is proposed, than with the present propelling system, all possibility of damage to the propellers due to solid objects being conveyed to the stern by the stream lines of water or to the propellers striking the quays or floating bodies alongside being completely obviated.

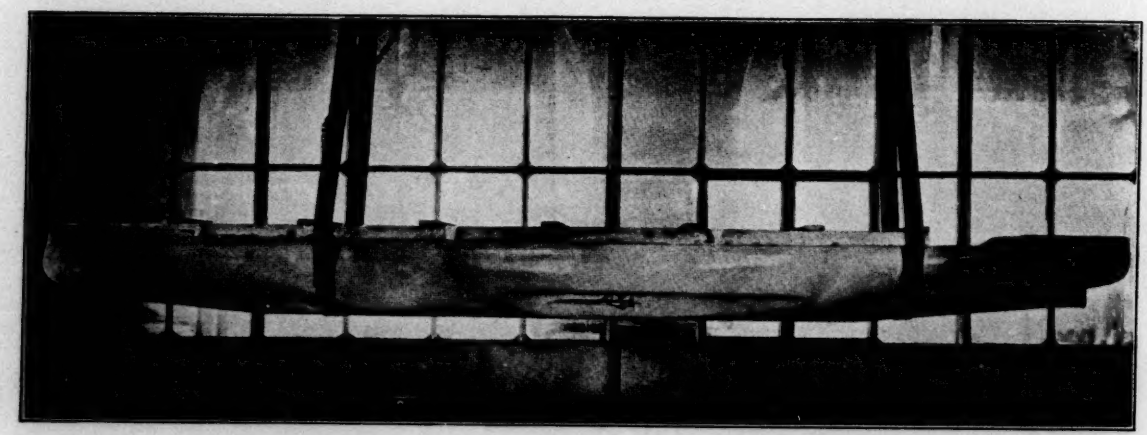
(7) That not only are the vibrations already referred to avoided, but a greater propulsive efficiency is obtained, and a vast improvement is effected in the steering and, incidentally, maneuvering of the vessel when using her propellers, giving a greatly reduced turning radius as the power of that movement is entirely concentrated amidships on an axis approximately at the center of gravity of the vessel. That a large saving in the cost of machinery will be effected by the adoption of the new system in every class of ship, which is a very important consideration, and that the new system is applicable to both mercantile vessels and warships, and is especially effective for high-speed passenger vessels driven with any class or type of machinery.

(8) That the steering of the ship by rudders is vastly improved owing to the avoidance of stream water from the propellers to the rudder.

(9) That in the case of vessels of very light draught for shallow waters the adoption of the new system of recesses in the hull, together with the new form of propeller, presents an improved feature. It solves the difficulty of adopting any existing form of screw propeller even in cases where the limitation due to the projected total surface of the blades of the propeller precludes the use of a screw propeller of present-day design, and where the limitation of the diameter of the screw arises out of the draught of the vessel.

(10) That one of the most important features of the new system is that the turning movement of the vessel is immensely increased, especially when she is entering harbor or maneuvering, as she turns almost on her own center with a radius of gyration which is practically nil, thereby obviating the necessity for the use of tugs, which is a special advantage in narrow channels.

The tendency of the day being towards increase in length and power of new ships the evils arising from vibration are necessarily intensified, and it is Mr. de Meo's opinion that the new system of central propulsion will prove to be indispensable in solving the problem of vibration of ships. We illustrate from photographs taken at the Froude Tank at Spezia the model on which the experiments were carried out with both a propeller of the standard design and the special propeller which Mr. de Meo proposes using, a separate view being given of the latter.—"Shipbuilding and Shipping Record," Oct. 1, 1925.



HULL MODEL WITH WHICH THE "DE MEO" PROPULSION TESTS WERE CARRIED OUT.

REPLACING AND REPAIRING PROPELLERS.

By Ben Shaw and James Edgar.

It is one thing to make a propeller from a good drawing and another to make one from a casting. For new vessels the patternmaker is supplied with a drawing on which is given an outline of the blade, the pitch, diameter, and sections at different radii from the center of the boss. If the patternmaker sets down the work correctly and makes the tackle with care, error is scarcely possible. The foundry work is also the patternmaker's responsibility, and he frequently not only sets the strickle on the spindle, but he measures the templets into position, dividing the mould face to do so. He subsequently locates the section pieces on the mould face which give the thickness of blade. It is very different when a new propeller has to be made to take the place of an old one. It is not proposed to describe here the patternwork and moulding, but the preparatory work before setting down the propeller is possible.

When the propeller of a vessel is damaged, the ship has to go into the graving dock, and the propeller has to be removed from the tailshaft. In most cases it is placed on the dock side, and a draughtsman or patternmaker takes particulars in order to have a new casting made. The casting has to be made in the shortest possible time, not only because the vessel may be due to sail in a few days, but also because of the high dock dues. Yet a propeller made urgently must be as accurate in the matter of pitch as one made for a new vessel.

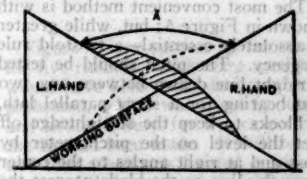


Fig. 1.

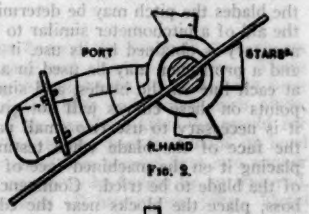


Fig. 2.

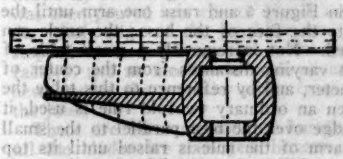


Fig. 3.

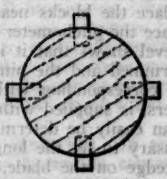


Fig. 4.

One of the most important points to determine is the hand of the propeller—that is, whether the blades form part of a right- or left-handed screw,—and it should be settled at the outset. It is wise, also, to determine the diameter and number of blades before any of the other details are noted. A method of determining the hand is illustrated in Figure 1. Right-angled triangles are formed upon a base-line as indicated, the base-line representing an imaginary circumference, and the height representing the pitch. Looking from the tip of the blade towards the boss, with the thickness of the blade as shown, a right-handed propeller would be represented, the top edges A representing the leading edge. When the hypotenuse of the other triangle represents the working surface of the propeller then it is required to be left-

handed. By making use of a sketch similar to Figure 1 any doubt about the correct hand will be quickly removed, and after a little practice the draughtsman will be able to visualize the sketch without special reference to it.

Measuring the pitch of an old propeller casting to insure accuracy in the new propeller is not an easy matter when one has had no experience. When the old propeller is near at hand, and the diameter of the small end of the bore can be obtained without unnecessary trouble, then a piece of wood can be cut to this diameter and a convenient parallel straightedge secured to it, as shown in Figures 2 and 3. When the old casting is not at hand, a smaller circular block may be prepared having four holes bored at the centers and about the periphery, so that short pieces of dowel stick may be knocked in each hole to make the diameter required, similar to that shown in Figure 4. Before lining off the old propeller it is necessary to have the casting laid as level as possible wtih the working faces of the blades upwards, as shown in Figure 2. With the aid of a spirit-level used on the machined end of the boss the casting can be chocked up to a suitable position for ascertaining the pitch. It is usually necessary to line off at least two blades so that one may check the other. From the center of the circular piece of wood attached to the straightedge, mark off a number of convenient distances to the tips of the blades. In large propellers these spaces are usually marked off in feet, but they may be varied to suit the size of the propeller. With the aid of a square and a straightedged lath the various distances can be projected to the blades by revolving the parallel straightedge. When assistance is not available, a weight attached to the short end of the straightedge to balance it will save a lot of trouble. Once the concentric lines have been marked on the blades the pitch may be determined. The most convenient method is with the aid of a pitchometer similar to that shown in Figure 5; but, while greater accuracy is obtained by its use, it is not absolutely essential—a twofold rule and a protracter may be used in an emergency. The pitch should be tested at each line on the blades, and since a straight line drawn between any two points on these curves will not give a flat bearing for a short parallel lath, it is necessary to use two small parallel blocks to keep the straightedge off the face of the blade when testing. Set the level on the pitchometer by placing it on the machined face of the boss and at right angles to the center of the blade to be tried. Commencing from the line on the blade nearest the boss, place the blocks near the edges and set the straight lath over them, then place the pitchometer as shown in Figure 5 and raise one arm until the spirit-level shows that it is parallel to the face of the boss; the angle can be determined and the pitch calculated. Not infrequently a small table of pitches, corresponding with angles at varying distances from the center of propellers, is supplied with a pitchometer, and by reference to this table the pitch can easily be determined. When an ordinary twofold rule is used, it is necessary to set the long straightedge over the boss parallel to the small straightedge on the blade, then one arm of the rule is raised until its top edge is considered to be out of winding with the long straightedge; the angle of the two arms to each other can then be determined with a protractor and the pitch calculated. Using the circumference of the line upon which the angle was determined as the base of a triangle, and knowing the angle between the base and the hypotenuse, the height or pitch can be found by multiplying the tangent of the angle with the base. In testing at a number of places over the surface of the blades any variance in pitch from the true screw will be duly noted. Once the pitch has been ascertained a center can be marked on a blade on a straight surface radiating into the shaft center; from this the varying sizes of the blade can be obtained and duly noted. When the pitch has been obtained a drawing may be made by the draughtsman, or the work can be set down on the shop drawingboard. The other dimensions are not difficult to obtain.

A possible alternative to making a new propeller when the tips have been damaged is to burn on new tips: a job which calls for much skill. The writers had some experience of burning propeller blade tips during the war, and were, therefore, specially interested in the burning of tips on a four-bladed propeller at the foundry of Negler & Co., of Antwerp. The propeller was 19 feet 6 inches diameter. The preliminary operation was to place the propeller with the face of the blade upwards on the foundry floor. Operations were then commenced on one blade, which was completed before a second blade was commenced. It would appear that a bed was made up underneath the outer end of the broken blades and was carried to beyond where the tip should have been. It is not clear whether a pattern was made for each tip or the mould was strickled. If it were strickled it would, of course, be necessary to draw an outline of the blade, carefully defining the leading and trailing edges.

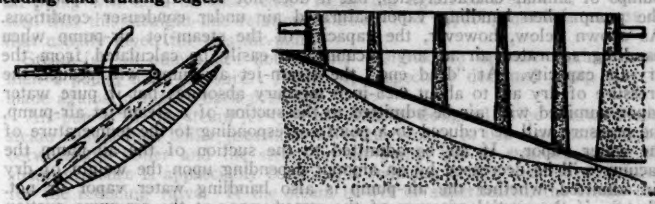


FIG. 5. REPLACING AND REPAIRING PROPELLERS. FIG. 6.

Figure 6 is a section of the mould just beyond the fracture, and shows the five runners which were used for pouring, and also the overflow. There would have been a big danger of cracking the old casting if it had not been preheated, which was done by kindling a coke fire underneath each blade, and letting it burn for five or six hours. This was done after the mould for the tip had been made. It will be noted that the top part was made up level and then a box was located on this. The metal was poured first into runner No. 1 until the edge of the old piece was shown to be fusing by an iron rod, which is, of course, the stereotyped test with all burning-on jobs. Metal was then poured into runner No. 2, and so on into each runner in turn until the whole edge of the propeller was fused, after which the overflow was stopped up and the tip cast. The job appeared to have been very satisfactory, and no mention is made of much dressing having to be done to the new part of the propeller apart from chipping off the runners and dressing the joint.

There are other methods of burning propeller blade tips. In most cases all the blades are damaged, because the damage is usually caused by the propeller, while revolving, striking something in the water. If, however, only one or two blades are damaged, the work of burning new tips is much simplified, because it is then possible to use the undamaged blade as a pattern. When this is done the propeller is packed up on a carriage, if it is not too large to go into a drying stove, with the faces of the blades upwards, or if it is too large to go into a drying stove it is rested on the foundry floor. It is always essential when burning to clean all the rust from the old casting, and it is also necessary to either chip the edge approximately straight or make a serrated edge.

It has usually been thought almost impossible to burn tips on manganese bronze propellers, chiefly, perhaps, because of the considerable contraction which always renders burning of non-ferrous castings more difficult than those of iron. For such propellers patterns should be made for each tip

required, these being left slightly thicker for subsequent finishing of the casting. Castings are then made from these patterns, which castings are packed up in correct relative position to the propeller and burnt on in the ordinary way. By this method the contraction difficulty is minimized. Some labor is naturally involved in finishing the new tips, but a correct screw is obtained. Needless to say, the danger of the old casting cracking is obviated by very careful preheating.—"Mechanical World," Sept. 11, 1925.

SELECTING THE RIGHT SIZE OF AIR-PUMP.

The capacity of an air-pump is usually expressed in cubic feet (atmospheric condition) of dry air per minute. This value is useful in comparing air-pumps of similar characteristics, but it does not state the true capacity of the pump when handling vapor-saturated air under condenser conditions. As shown below, however, the capacity of the steam-jet air-pump when handling saturated air at any vacuum can easily be calculated from the dry-air capacity. At "dead end" the steam-jet air-pump will reduce the pressure of dry air to about 0.25-inch mercury absolute; but if pure water vapor unmixed with air be admitted to the suction of a steam-jet air-pump, the pressure will be reduced to a point corresponding to the temperature of the water vapor. If air be admitted to the suction of the air-pump the vacuum will be decreased by an amount depending upon the weight of dry air admitted, whether the air-pump is also handling water vapor or not. That is, if the partial pressure of the water vapor at the air-pump suction

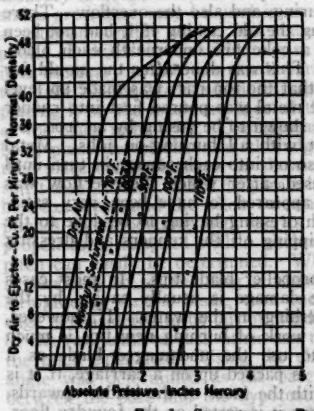


FIG. 1. SELECTING THE RIGHT SIZE OF AIR-PUMP.

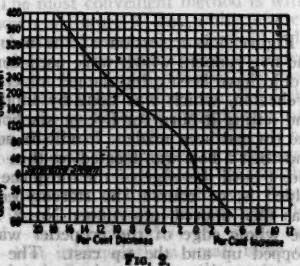


FIG. 2.

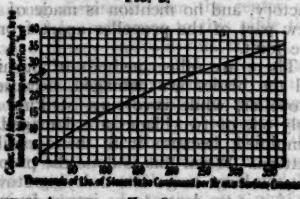


FIG. 3.

remains the same, a given quantity of air will decrease the vacuum by the same amount, no matter whether the temperature of the water vapor is 80 degrees or 110 degrees.

Thus, if it is assumed that 26 cubic feet (normal density) of dry air leaks into the condenser in one minute, and that it must be removed saturated with water vapor at 90 degrees F., the vacuum which a certain size steam-jet

air-pump will produce can be determined as follows: From the dry-air test it is known that at dead end the air-pump will produce a 0.29-inch vacuum, and that if 26 cubic feet of dry air be admitted per minute, the vacuum will fall 0.71 inch to an absolute pressure of 1 inch. Saturated water vapor at 90 degrees F. has an absolute pressure of 1.41 inches, so the pressure in the condenser with an inleakage of 26 cubic feet of dry air per minute will be 1.41 inches + 0.71 inch, or 2.12 inches of mercury. Examining Figure 1, it will be noted that the vacuum actually produced under these conditions is 2.16 inches of mercury. Expressed algebraically, $P = Pw + (Pa - Po)$, where P is equal to the absolute pressure produced in the condenser, Pw the pressure of saturated water vapor at the temperature of the air-pump suction, Pa the absolute pressure produced by the steam jet when handling the same weight of dry air on orifice test, and Po the pressure which the air-pump will hold on dead end. Referring again to Figure 1, it can be seen that this straight-line relation holds up to a capacity much in excess of that usually given for this air-pump.

Steam-jet air-pumps are not usually designed to operate against more than 1 pound back pressure. However, a specially designed pump may be made against a higher back pressure.

Air-pumps of the steam-jet type are usually designed to be operated with steam at less than boiler pressure. It is customary to use a reducting valve between the boiler and the steam-jet air-pump, but in stations where the steam pressure is appreciably constant, a reducing orifice may be substituted for the reducing valve. If the steam is wet or superheated, the steam consumption will vary, as shown in Figure 2, being decreased with superheated steam and increased with wet steam.

The method of determining the size of air-pump suitable for a given condenser is largely empirical, as the air inleakage is variable, depending upon the care taken to keep the turbine and condenser joints in good condition. The number of pounds of steam condensed per hour at full load is generally used as a basis for calculation (in spite of the fact that little air enters directly with the steam), on the assumption that the size of the condenser, and therefore the chance of air inleakage, is proportional to the amount of steam to be condensed. The curve in Figure 3 gives a safe value of the air-handling capacity to be installed with condensers of various sizes. The capacity in cubic feet of atmospheric air per minute of the steam-jet air-pump as shown by a dry-air test is plotted against the pounds of steam to be condensed per hour in the surface condenser. That air-pump should be chosen which will have the desired free air capacity at the vacuum which it is expected to obtain. In other words, says a writer in "Power Plant Engineering," if the temperature of the cooling water is such that the highest vacuum possible is 28 inches, the air-pump need only have the required free-air capacity at 28 inches, whereas if 29-inch vacuum is obtainable, a larger air-pump with the required capacity at 29 inches should be chosen. For jet condensers an air-pump twice as large is usually installed.—"Mechanical World," Sept. 11, 1925.

BOOK REVIEWS.

A GRAPHIC TABLE COMBINING LOGARITHMS AND ANTI-LOGARITHMS BY LACROIX AND RAGOT. THE MACMILLAN COMPANY, NEW YORK.

This book is an improvement on the usual tables and consists of forty pages of five-place graphic tables and six pages of four-place tables. The logarithms of all five-place numbers can be read directly from the scale without the necessity of interpolation; if greater accuracy is desired, the sixth-place can be interpolated from the divisions of the scale. There is less chance for errors, as it is not necessary to interpolate in the case of any number of five figures. The book is compact, printed on good paper, handy in size, and will undoubtedly be preferred by many over the ordinary tables.

"MAKERS OF NAVAL TRADITION," BY ALDEN AND EARLE. GINN AND COMPANY, 15 ASHBURTON PLACE, BOSTON 2.

A very interesting book by authors in good position to gain complete knowledge of the subject. It is written as a biography of some of the best known officers of the Navy who, with their contemporaries, built the traditions of the Navy, among them Jones, Decatur, Macdonough, M. C. Perry, Maury, Dahlgren, Farragut, Porter, Luce, Mahan, Dewey and Sampson. It should be especially inspiring to those in the service and of interest to all students of our Naval History.

ASSOCIATION NOTES.

At a meeting of the Council of the Society, held on September 22, 1925, the resignation of Rear Admiral C. F. Hughes was accepted due to his detachment from duty in Washington to assume command of the Battle Fleet. In accordance with the provisions of the By-Laws Rear Admiral Montgomery M. Taylor was appointed President by the Council to serve the unexpired term of Admiral Hughes.

The annual business meeting of the Society was held on Tuesday, October 6, 1925 at Washington, D. C. Nominations were made for officers of the Society for the year 1926 as follows:

For *President:*
Rear Admiral M. M. Taylor, U. S. N.

For *Secretary-Treasurer:*
Commander A. M. Charlton, U. S. N.

For *Member of Council:*
Rear Admiral C. W. Dyson, U. S. N.
Captain W. T. Cluverius, U. S. N.
Captain Q. B. Newman, U. S. C. G.
Captain S. M. Robinson, U. S. N.
Commander E. E. Wilson, U. S. N.
Lieutenant Commander E. L. Cochrane (CC) U. S. N.
Mr. J. F. Metten.
Mr. J. F. Nichols.
Mr. H. M. Southgate.

Ballots have been mailed to all members entitled to vote.
It was voted to hold the usual annual banquet in 1926.

The following have joined the Society since the publication of the August 1925 JOURNAL:

NAVAL MEMBERS.

Bailey, Carlos A., Lieutenant Commander, U. S. N.

Finn, W. A., Lieutenant, U. S. N.

Laws, George W., Rear Admiral, U. S. N.

Leigh, R. H., Rear Admiral, U. S. N.

Tilley, Benjamin F., Lieutenant Commander, U. S. N.

CIVIL MEMBERS.

Magdeburger, Edward C., 4419 New Hampshire Avenue, Washington, D. C.

ASSOCIATE MEMBERS.

Davis, John A., Naval Station, Pearl Harbor, T. H., 3263 Nuuaun Ave., Honolulu.

Lin, Chang S., Fort DuPont, Del.

Morton, John W., 532 N. 19th St., Philadelphia, Pa.

Sjöström, J. A., Kockums Mek, Verkstads A. B., Malmo, Sweden.

The following have been transferred from

ASSOCIATE TO CIVIL MEMBERSHIP.

Burkhardt, John E., Fore River Plant, Bethlehem Shipbuilding Corporation, Quincy, Mass.

Wetherbee, Charles P., 110 High St., Bath, Me.